ENGINE AND BOILER DESIGN

HOFFMAN

A COURSE OF INSTRUCTION

IN

ENGINE AND BOILER DESIGN

ARRANGED FOR

STUDENTS OF THE SENIOR CLASS
PURDUE UNIVERSITY
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BY

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STEAM ENGINE DESIGN

REFERENCE LIST.

Young—Steam Engine and Boiler Notes and Problems.

Hutton—Mechanical Engineering of Power Plants.

Whitham—Steam Engine Design.

Klein—Steam Engine Design.

Rigg—Practical Treatise on Steam Engines.

Rankine—Steam Engines.

Barr—Current Practice in Engine Proportions.

Cotterill—The Steam Engine.

Unwin—Elements of Machine Design, Vol. II.

Thurston—A Manual of the Steam Engine.

Reuleaux—The Constructor.

Holmes—The Steam Engine.

Kent—Mechanical Engineers' Pocket Book.

In connection with the above, look up the numbers of the Engineering Index; they come monthly and give references from over two hundred engineering papers. Three bound volumes of this index have appeared, the last one in 1902. These volumes contain a vast amount of information concerning references on the design of steam engines.

In the following notes, reference is frequently made to the "Manufacturers Average" (M. A.). These references are taken from a thesis presented in 1904 by Mr. F. A. Berger on "The Comparison and Development of Formulae for the Design of High Speed Steam Engine Details." This investigation has not only summed up existing formulas, but has averaged the standard sizes of the parts in engines of the various types.

PROBLEMS IN HIGH-SPEED STEAM ENGINE DESIGN.

In developing a design of this kind it is not required that each man keep an accurate record of all the work on the individual parts made by the other designers. It will be advisable, however, to keep in as close touch as possible with their work. To accomplish this, the following problems should be worked out by every member on the design and submitted, with such other notes and remarks on the design as he deems important, at the end of the time when the design shall be finished and at such other times during the progress of the work as they shall be called for by the instructor.

Sizes shall be calculated where possible and wherever such calculated sizes are departed from, both calculated and accepted sizes should be given. It is urged that frequent reference be made to late catalogues and other information to check up the work with current practice.

PROBLEM 1. Determine the size (diameter and stroke) of the engine cylinder for your assigned horse power.

PROBLEM 2. Estimate the horse power of the engine at 1-10, 1-4 and 7-10 cut-off, using the planimeter for the M. E. P.

PROBLEM 3. Estimate the weight of the reciprocating parts by Barr's formula.

PROBLEM 4. Estimate the force necessary to accelerate the reciprocating parts with the engine on first, head end dead center; second, crank end dead center.

PROBLEM 5. Determine the coefficient of fluctuation of energy for 1-10, 1-4 and 7-10 cut-off.

PROBLEM 6. Estimate the number of foot pounds of energy that must be taken up and given out by the flywheel at 1-10, 1-4 and 7-10 cut-off.

PROBLEM 7. In calculating the flywheel for this engine estimate the width of belt,

(a), if single belt is used.

(b), if double belt is used.

State size of belt adopted and give width of face of flywheel.

PROBLEM 8. Determine the weight of the flywheel and the thickness of the rim.

PROBLEM 9. Determine the diameter of the crank shaft.

PROBLEM 10. Determine the dimensions of the crank pin; projected area, length and diameter.

PROBLEM 11. Determine the dimensions for the wrist pin; projected area, length and diameter.

PROBLEM 12. Determine the sections of the crank arm at the pin and at the center of the shaft.

PROBLEM 13. Determine the section of the connecting rod at the wrist pin end, regarding the rod as in tension.

PROBLEM 14. Estimate the ratio of the given load on the connecting rod to that load that would start flexure considering the rod as a column.

PROBLEM 15. Determine the area of the bearing surface of the cross head.

PROBLEM 16. Determine the diameter of the piston rod.

PROBLEM 17. Estimate the ratio of the given load on the piston rod to the load that would start flexure, assuming the rod to be pin and square ended. Assume also $l = \frac{3}{2}$ length of stroke.

PROBLEM 18. Determine the width of the face of the piston.

PROBLEM 19. Determine the thickness of the cylinder wall; also that of the cylinder head.

PROBLEM 20. Determine the number and diameter of studs for the cylinder head.

PROBLEM 21. Determine the following steam passages:

(a), steam pipe; area and diameter.

(b), exhaust pipe; area and diameter.

(c), cylinder ports; area, length and width.

NOTES ON THE DESIGN OF HIGH SPEED STEAM ENGINES.

The terms *low speed* and *high speed* as applied to steam engines relate to the *revolutions per minute.* It is usually understood that low speed engines have from 100 to 200 R. P. M. and high speed engines from 200 to 350 R. P. M.

Piston Speed:—The piston speeds of stationary engines vary from 500 to 700 F. P. M. The lower speeds are found on the smaller engines, say from 10 to 30 H. P. The piston speeds are about the same on low and high speed engines of the larger sizes, except where designed for special work; for illustration: a 13″ x 15″ high speed engine runs at about 250 R. P. M., and an 18″ x 36″ Corliss runs at about 100 R. P. M. each having approximately 600 F. P. M., piston speed. The average of a number of actual cases, however, shows a speed of 585 F. P. M.

Revolutions Per Minute:—The average of a number of engines shows the following: 50 H. P., 290 R. P. M.; 100 H. P., 260 R. P. M.; 150 H. P., 230 R. P. M.; a decrease of 30 R. P. M. for each increase of 50 H. P.

Theoretical Indicator Card:—In beginning the design of a steam engine the first thing to be considered is the theoretical indicator card. In order to obtain this a few assumptions must be made, namely: boiler-pressure, cut-off, compression, admission, release, clearance and back pressure.

Boiler Pressure:—This ranges in practice from 85 to 125 pounds gauge pressure. Very high steam pressures are not economical in single cylinder engines, but are used in multiple cylinders and condensing engines. Current practice seems to favor 90, 95 and 100 pounds gauge for the 50, 100 and 150 H. P. single cylinder non-condensing engines, respectively.

Cut-Off:—Cut-off varies from 10% to 70% of the stroke, the normal rating being at 25%. It will be well in this work to assume three cut-offs, 10, 25 and 70%, so as to permit of satisfactory comparison.

Compression:—This serves two purposes, to heat up the cylinder and to cushion the reciprocating parts. We will not consider theoretical values here, but will take an average obtained from an analysis of a number of engines, as follows:—

10% cut off......45% Compression.
25% cut off......34% Compression.
70% cut off......12% Compression.

Admission and Release:—Admission and release depend upon cut-off and compression and can not be arbitrarily assumed. The following is an average taken from the analysis previously mentioned.

10% cut-off.	8% admission	56% release.
25% cut-off	3% admission	67% release.
70% cut-off	0% admission	92% release.

It seems scarcely necessary to make any assumption regarding admission and release on the theoretical cards, since no values could be assigned to fulfil the average conditions as indicated for the three cut-offs. Some designers take admission and release at the ends of the stroke. In either case errors will be involved, but it is altogether probable that the errors will be less when admission and release are considered as given in the table.

Clearance:—Clearance may be divided into two parts: (a), between the piston and cylinder head, this varies from 1/8" to 3/8" for rough castings, and $\frac{1}{16}''$ to $\frac{1}{8}''$ for finished castings; (b), steam passage. Total clearance will vary from 6 to 12% of the steam volume per stroke in high speed engines. This may be taken at 8%. Low speed engines will have much less percentage of clearance, some times as low as 2% of the steam volume per stroke.

Back Pressure:—Assume back pressure as follows:

Non-condensing 2 to 4 pounds gauge; 17 to 19 pounds absolute.

Condensing 2 to 4 pounds absolute.

Cylinder Size:—Having been assigned the horse power of the engine to be designed, the first operation is to find the diameter of the cylinder from the formula:

$$\text{H. P.} = \frac{P\,V}{33000} \tag{1}$$

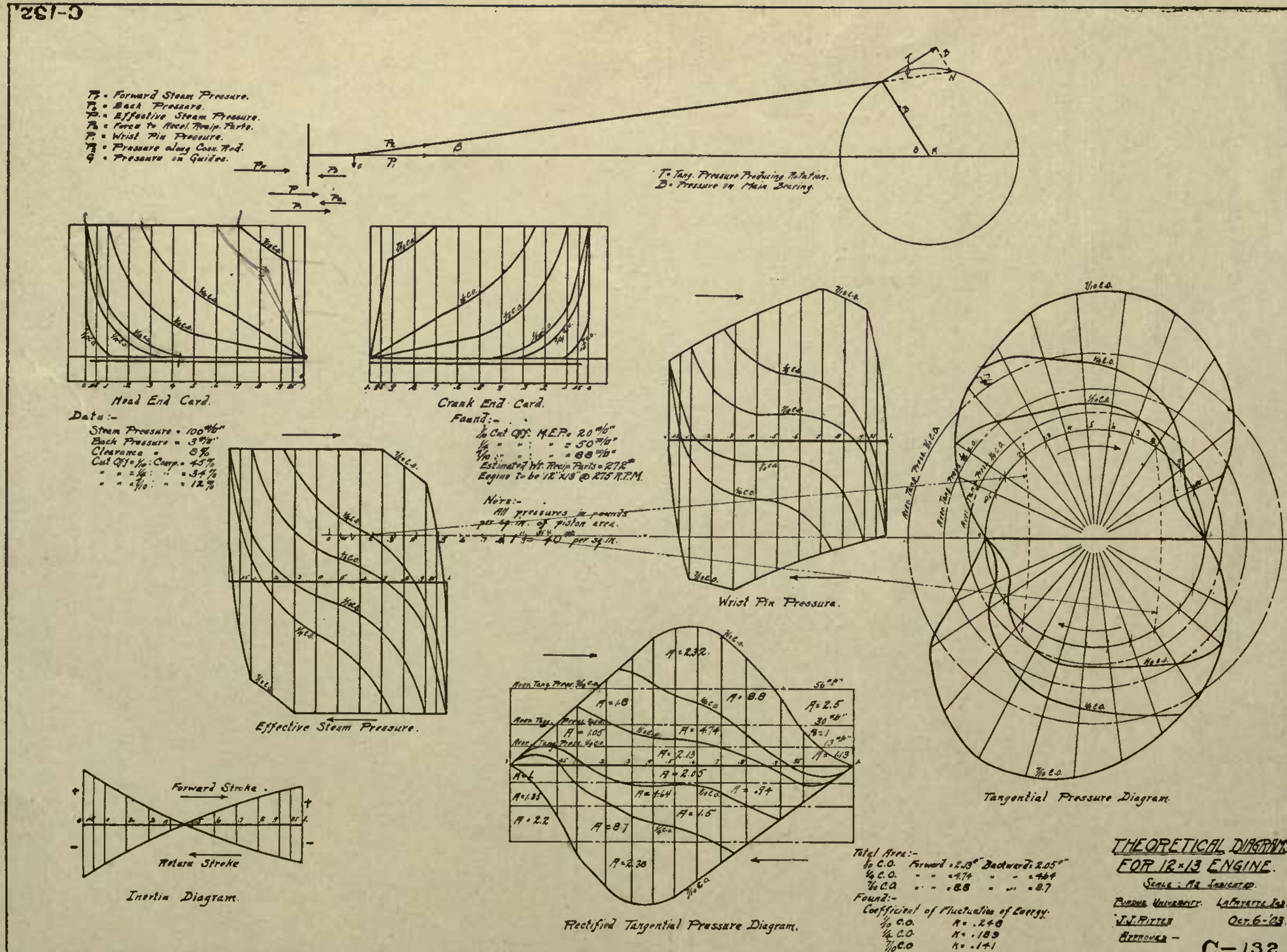
where P = M. E. P. $\times$ Area of piston and V = Piston speed in F. P. M.

Find M. E. P. from the formula (2) and substitute in (1)

$$\text{M. E. P.} = p\,\frac{(1 + hyp.\ log.\ r)}{r} - \text{back pressure.} \tag{2}$$

where p = absolute initial pressure, and r = number of expansions. In obtaining r, it will not be necessary to consider clearance.

Note.—Formula (2) is an approximate formula which gives results entirely satisfactory for the determination of the cylinder diameter. After the theoretical cards have been drawn the areas should be taken with a planimeter and the M. E. P. for the three cut-offs computed from these areas. It will be of interest, then, to compare the results obtained in the two ways.



C-132.
P_1 = Forward Steam Pressure.
P_2 = Back Pressure.
P_3 = Effective Steam Pressure.
P_4 = Force to Accel. Recip. Parts.
P = Wrist Pin Pressure.
P_5 = Pressure along Conn. Rod.
G = Pressure on Guides.
T = Tang. Pressure Producing Rotation.
B = Pressure on Main Bearing.
Head End Card.
Crank End Card.
Data:-
Steam Pressure = 100 lbs.
Back Pressure = 3 lbs.
Clearance = 8%
Cut Off = 1/10 : Comp. = 45%
" = 1/4 : " = 34%
" = 1/2 : " = 12%
Found:-
1/2 Cut Off. M.E.P. = 20 lbs.
1/4 " " = 50 lbs.
1/10 " " = 88 lbs.
Estimated Wt. Recip. Parts = 275 lbs.
Engine to be 12"x13" @ 275 R.P.M.
Note:-
All pressures in pounds per sq. in. of piston area.
Effective Steam Pressure.
Wrist Pin Pressure.
Tangential Pressure Diagram.
Inertia Diagram.
Forward Stroke.
Return Stroke
Rectified Tangential Pressure Diagram.
Total Area:-
1/2 C.O. Forward = 2.13" Backward = 2.05"
1/4 C.O. " = 4.74 " = 4.64
1/10 C.O. " = 8.8 " = 8.7
Found:-
Coefficient of Fluctuation of Energy.
1/2 C.O. K = .248
1/4 C.O. K = .189
1/10 C.O. K = .141
THEORETICAL DIAGRAMS
FOR 12 x 13 ENGINE.
Scale: As Indicated.
Purdue University. Lafayette, Ind.
J.J. Ritter Oct. 6-'03.
Approved -
C-132.

Another approximate rule for finding the cylinder diameter is $H. P. = \frac{3}{4} D^2$ (3)

On the basis of the manufacturers' rating this formula would become $H. P. = \frac{2}{3} D^2$. This rating is, however, below the average engine performance.

After obtaining the cylinder diameter D, assume the length of stroke L, and the number of revolutions per minute N, such that $L\ N =$ Piston speed in F. P. M.

Manufacturers' average: $L = 1.122\ D =$ *say* $1\frac{1}{8}\ D$.

In selecting the cylinder sizes the following ratios may be of value.

7", 8", 9", 10" and 11" x 10"
10", 11", 12", 13" and 14" x 12"
12", 13", 14", 15" and 16" x 14"
11", 12", 13", 14" and 15" x 15"
14", 15", 16", 17" and 18" x 16"

Theoretical Diagrams Sheet:—In planning this sheet, Plate C — 132 may be used as a guide. Draw the Force Diagram at the top to any convenient scale, then draw the *Theoretical Indicator Cards* for each of the three cut-offs and combine the forward and backward cards, simultaneously produced, into the *Effective Steam Pressure Cards*.

To obtain the diagram representing the *inertia of the reciprocating parts*, i. e., piston, piston rod, cross-head and one half of the connecting rod, it will be necessary to estimate the weight of these parts. This is commonly done by Barr's formula:

$$W = 1{,}860{,}000\ \frac{D^2}{L\ N^2} \qquad (4)$$

where $W =$ weight in pounds, $D =$ diameter of cylinder in inches, $L =$ length of stroke in inches, N = R. P. M.

The M. A. seems to run somewhat greater than the above and is represented by the formula:

$$W = \frac{1{,}900{,}000\ D}{N^2}$$

After the value of W is obtained it is then substituted in the formula:

$$F = \frac{W v^2}{g\ r}\left[\cos\theta \mp \frac{\sin^2\theta}{\sqrt{\frac{l^2}{r^2} - \sin^2\theta}} \pm \frac{l^2}{r^2}\,\frac{\cos^2\theta}{\left(\frac{l^2}{r^2} - \sin^2\theta\right)^{\frac{3}{2}}}\right] \qquad (5)$$

where $F =$ force necessary to accelerate the reciprocating parts, $W =$ weight of reciprocating parts in pounds, $v =$ velocity of crank pin in F. P. S., $l =$ length of connecting rod in feet, $r =$ Radius of crank in feet, θ = Crank angle with the horizontal, $g = 32.2$.

$\frac{r}{l}$ may be taken from $\frac{1}{4}$ to $\frac{1}{6}$.

In applying (5) it will be necessary to solve for $\theta = 0°$ and $\theta = 180°$.

For $\theta = 0°$ the acceleration is negative and falls below the horizontal line, for $\theta = 180°$ the acceleration is positive and is shown above the line. Since the inertia card on the return stroke is of the same size, representing no loss of energy, the positive and negative forces become negative and positive, respectively, as shown. This gives two points in the inertia curve, i. e., the extremes. The formula can then be solved for intermediate values of θ between 0° and 180° and an accurate curve produced; this, however, has been found to approximate to the arc of a circle passing through the extremes as found, and p, a point on the x, axis where the acceleration passes from negative to positive and vice versa. This point p will lie at

.46 stroke when $r \div l = \frac{1}{6}$
.45 stroke when $r \div l = \frac{1}{5}$
.44 stroke when $r \div l = \frac{1}{4}$

After locating the points as stated, pass arcs of circles through them and these arcs will represent the acceleration curves.

The curves representing the *forces on the wrist pin* are next plotted by combining the effective steam pressure cards and the inertia cards. It will be noticed that the effect of the inertia card is to round up the final card and more nearly equalize the effective forward forces.

The forces representing the *tangential pressures on the crank pin* are obtained by multiplying the forward forces on the wrist pin by the values $A\ N \div r$ given in the following table. Concerning $A\ N \div r$ see "Stahl & Wood," Elementary Mechanism, Page 196; also Steam Engine & Boiler Notes (Young.)

Values of $A N \div r$.

Piston Position	$r \div l = \frac{1}{6}$	$= \frac{1}{5}$	$= \frac{1}{4}$
.00	.00	.00	.00
.05	.47	.48	.50
.10	.64	.65	.67
.20	.84	.86	.88
.30	.95	.96	.98
.40	1.01	1.02	1.03
.50	1.01	1.01	1.02
.60	.97	.97	.96
.70	.88	.87	.85
.80	.76	.74	.72
.90	.54	.52	.50
.95	.39	.36	.36
1.00	.00	.00	.00

For any other ratio of $r \div l$ the value $A N \div r$ can be found either analytically or graphically. Locate the corresponding positions of the crank pin on the crank circle, .05, .1, .2, .3 etc., draw indefinite radial lines through them, lay off these products outside the crank circle and through the points obtained draw the irregular curve representing the *locus* of the *tangential forces* exerted on the crank pin.

The *rectified tangential* curves are obtained by drawing the x axis, the length of the half circumference of the crank circle, and at the points, .05, ,1, .2, .3 etc., erect perpendiculars of equal length to the radial lines outside the circle in the tangential pressure diagram and through these extremities draw the rectified arc. Find the area of each half of the enclosed figure and divide by the length. This will give the average height of card, or the *mean tangential pressure line.* See also Whitham, page 199; Heck, Notes Supplementing Holmes, Page 23; Holmes, Page 196; Unwin, Page 173; Marks, Page 120; Klein, Page 63.

Fly Wheel:—The lack of uniformity in the turning effort at the crank, produces fluctuations of speed in the engine. The way in which this may be largely overcome is to design a fly wheel of sufficient weight to absorb and control these fluctuations. In the steam engine notes the formula for the weight of the fly wheel is given as

$$W = \frac{K E G}{n\, v^2} \qquad (6)$$

where

W = weight of the wheel in pounds.

$$K = \frac{b\, c\, d}{p\, q\, q'\, p'} \text{ or } \frac{d\, q\, q'\, d'}{p\, q\, q'\, p'} \; *$$

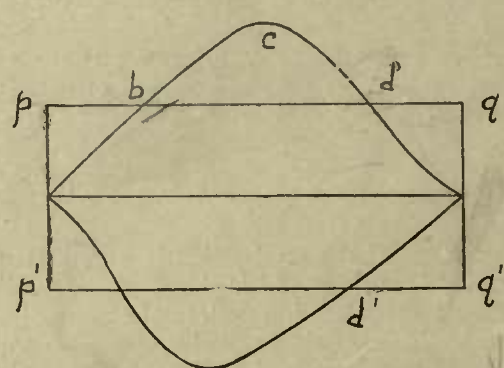


Fig. 1.

E = energy of one revolution in foot pounds

G = 32.2

n = coefficient of fluctuation of speed, taken at 1-100 to 1-150 for high speed engines. For electric lighting the higher figure would be used.

v = velocity of the fly wheel rim in F. P. S. The velocity of the fly whel rim would be the same as that assumed for the belt which may be taken at 4,000 F. P. M. Practice varies on this point from 3,000 to 6,000 F. P. M. The maximum efficiency of a belt is obtained in practice at about 5,000 F. P. M., but the ordinary practice for high speed work is about as stated, with smaller engines say 30 to 50 H. P., some below; and larger ones, say 100 to 150 H. P., some above this figure.

*These two areas are not equal, $b\ c\ d$ is commonly used. The suggestion is made that the greatest fluctuation should be taken and if the two deficiencies $d\ q\ q'\ d'$ be greater than $b\ c\ d$, it should be used instead.

In selecting v it would be well to consider what is the best diameter for the wheel. The (M. A.) gives

$$D = .0235 \sqrt{H.P. \times N} \tag{7}$$

where N = R. P. M. and D = diameter in feet.

The assumed velocity can then be checked up with the above before substituting it in the formula for the weight of the wheel.

The weight, as found, would naturally be that of the entire wheel as though it were all fixed in the rim. This is not theoretically correct since the arms have considerable weight, but in practice it is generally taken in this way.

Other Formulas for Weight of Fly Wheel.

Barr:—$W = 833{,}000{,}000{,}000 \dfrac{H.P.}{D^2 N^3}$ $\left\{ \begin{array}{l} D = \text{Diameter of wheel in inches.} \\ N = \text{R. P. M.} \end{array} \right.$

Thurston:—$W = 960{,}000{,}000{,}000 \dfrac{H.P}{D^2 N^3}$ $\left\{ \begin{array}{l} D = \text{Diameter of wheel in inches.} \\ N = \text{R. P. M.} \\ \text{where M. E. P. is not less than 30.} \end{array} \right.$

Stanwood:—$W = 1{,}000{,}000 \dfrac{D^2 L}{D_1^2 N^2}$ $\left\{ \begin{array}{l} D = \text{diameter of cylinder in inches.} \\ L = \text{stroke in inches.} \\ D_1 = \text{diameter of wheel in feet.} \\ N = \text{R. P. M.} \end{array} \right.$

Whitham:—$W = \dfrac{387{,}587{,}500\, K\, n \times H.P.}{D^2 N^3}$ $\left\{ \begin{array}{l} n = \text{variation of speed say 1-50 to 1-100.} \\ D = \text{diameter of rim in feet.} \\ N = \text{R. P. M.} \end{array} \right.$

$$K = \frac{\text{excess of turning effort over resistance}}{\text{whole crank effort during one stroke.}}$$

Klein:—$W = 388{,}000{,}000 \dfrac{K \times H.P.}{f D^2 N^3}$ $\left\{ \begin{array}{l} f = \text{coefficient of unsteadiness} = \text{1-50 to 1-100.} \\ D = \text{diameter of wheel in feet.} \\ N = \text{R. P. M.} \end{array} \right.$

$$K = \frac{\text{excess of power or resistance during any phase}}{\text{total power exerted during one revolution.}}$$

(M. A.): — $W = H.P.\ (.05\ N + 10)$ N = R. P. M.

With a given velocity of the belt and an assumed working strength of the belt per square inch of section, the next operation is to decide on either single, double or triple belt and solve for the width.

Concerning the working strength of the belt much might be said, and many references might be cited, chiefly those in Kent, Pages 876-887. These references and others that might be given show such a lack of uniformity that it becomes largely a matter of the judgment of the designer. Tests of belting show an ultimate strength of about 4,000#□″; allowing a factor of safety of 10, we would have 400#□″ as a maximum working strength. This figure is, however, seldom reached. The extremes of practice for the working strength per square inch of section are about 250 to 400, which agrees in practice with a tractive force of from 180 to 290 pounds per square inch of section. Kent seems to favor a tractive force of 275 pounds per square inch of section which is equivalent in a double belt to about 86 pounds per inch of width. This figure seems to give results that agree well with current practice in high speed engine work. Concerning the thickness of the belt it may be said that for power work on engines exceeding 50 H. P. a double belt would be preferred. The following thicknesses may be accepted for belting:—

single belt..............3-16″
double belt.............5-16″
triple belt..............7-16″

Taking the formula $P\ V \div 33000$ = H. P. we can obtain the total tractive pull P on the belt, then knowing the tractive force of the belt per inch in width, the width in inches of belt can be obtained. Having the width of the belt, the width of the pulley face is obtained by adding from ½ to 2 inches to the belt width.

Another very satisfactory formula, if it is desired to take into account the centrifugal tension and the arc of contact of the belt, is that given in Kent, Page 878, by Nagle.

$$\text{H. P.} = C\,V\,t\,w\,\frac{(f - .012\,V^2)}{550}$$

where $C = 1 - 10^{-.00758\,\phi\,a}$

a = degrees of belt contact.

Φ = coefficient of friction generally taken at from .3 to .4

w = width of belt in inches.

t = thickness of belt in inches.

V = velocity in feet per second.

f = stress on belt per square inch of section.

Take

f = 275 for laced belts.

f = 400 for lapped and rivetted belts.

Table of Values of $C = 1 - 10^{-.00758\,\phi\,a}$

For different arcs of contact.

Φ = Coeff of Friction.	Degrees of Contact = a										
	90	100	110	120	130	140	150	160	170	180	200
15	.210	.230	.250	.270	.288	.307	.329	.342	.359	.376	.408
20	.270	.295	.318	.342	.364	.386	.408	.428	.448	.467	.503
25	.325	.354	.381	.407	.432	.457	.480	.503	.524	.544	.582
30	.376	.408	.438	.467	.494	.520	.544	.567	.590	.610	.649
35	.423	.457	.489	.520	.548	.575	.600	.624	.646	.667	.705
40	.467	.502	.536	.567	.597	.624	.649	.673	.695	.715	.753
45	.507	.544	.579	.610	.640	.667	.692	.715	.737	.757	.792
55	.572	.617	.652	.684	.713	.738	.763	.785	.805	.822	.853
60	.610	.649	.684	.715	.744	.769	.792	.713	.832	.848	.877
100	.792	.825	.853	.877	.897	.913	.927	.937	.947	.956	.969

The (M. A.) for width of fly wheel pulley rims is $W = 1.35\,\sqrt{\text{H. P.}}$

In determining the thickness of the pulley rim having given the width, the following formula will be sufficiently accurate.

$$W = .26\,\pi\,w\,(D - t)\,t$$

where D = diameter of pulley in inches.
t = thickness of rim in inches.
W = weight of pulley in pounds.
w = width of rim in inches.
(M. A.). $t = .27\,D + .5''$ where D = diameter in feet.

Some standard pulley sizes:—

Diameter in inches	Face in inches.
36″	6½″.... 8½
42″	8½″....10½
48″	8½″....12½
54″	10½″....12½
60″	10½″....14½
66″	12½″....14½
72″	12½″....14½

The shape of the arm section is then selected and the arm calculated as a beam under flexure. If this section is oval as shown in the figure, the formula becomes

$$\frac{P\,L}{N} = .05\,b^3 f$$

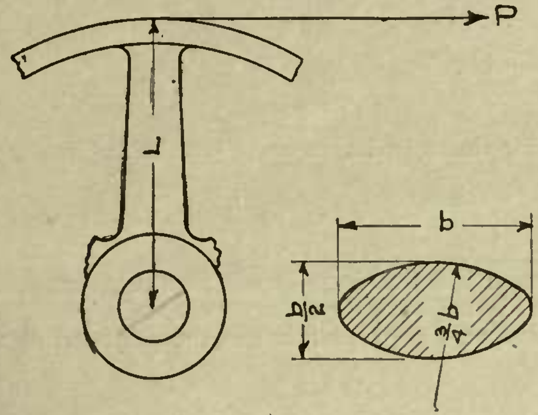


FIG. 2.

where P = Tractive force of belt

f = Allowable fibre stress of the metal, say 2000#□″;

N = Number of arms.

b = Breadth of arm at center of wheel if projected to that point.

The breadth and thickness of the arm at the rim may be taken at approximately 2-3 of the hub sizes.

For the diameter of the hub a very common practice is to take a value equal to twice the diameter of the shaft. (= 2 d)

For the size of the key use:—

$$w = \begin{cases} \dfrac{d}{5} \text{ for a six inch shaft to} \\ \dfrac{d}{4} \text{ for a two inch shaft.} \end{cases}$$

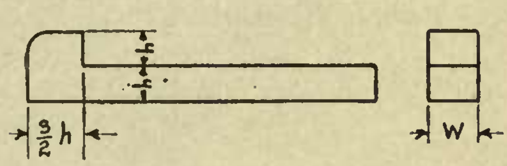


FIG. 3.

$h = \left(\frac{2}{3} w \text{ to } \frac{3}{4} w\right)$ for a six inch shaft $\qquad h = \left(\frac{3}{4} w \text{ to } w\right)$ for a two inch shaft.

Bed:—No rules can be laid down concerning the weight of the engine bed. It seems to vary between 20 and 35 pounds per horse power.

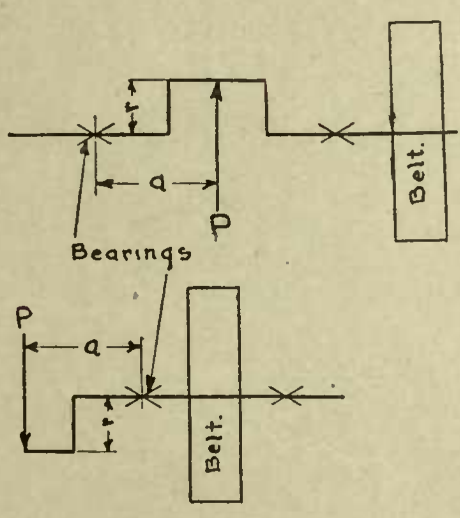


FIG. 4.

Shaft:—In every crank shaft, no matter if the engine is side crank or center crank, two forces are acting: (1) A twisting movement T due to $P\,r$, (2), A bending movement M due to $P\,a$. The two can be reduced to an "equivalent twisting movement T' as shown by Low & Bevis, Page 95, and Whitham, Page 245.

$$T' = M + \sqrt{M^2 + T^2} \qquad (9)$$

then find the diameter from

$$d = 1.72\sqrt[3]{\frac{T'}{f}} \qquad (10)$$

A good value for f in such cases is 8,000.

The above gives safe values except with very heavy fly wheels where the bending due to the weight of the fly wheel must be taken into account.

Other formulas are given as follows:

$$\text{Kent:—}\begin{cases} d = .43\,D \text{ for long stroke engines.} \\ d = .40\,D \text{ for short stroke engines} \end{cases} D = \text{diameter of cyl. in inches.}$$

I. C. S.:—$d = .44\,D + \frac{1}{2}''$ . . . . D = diameter of cyl. in inches.

$$\text{Barr;} - \begin{cases} d = 7.3 \sqrt[3]{\dfrac{H.\ P.}{R.\ P.\ M.}} & \text{for high speed engines.} \\ d = 6.8 \sqrt[3]{\dfrac{H.\ P.}{R.\ P.\ M.}} & \text{for low speed engines.} \end{cases} \qquad (11)$$

(M. A.) :—$d = .0053\ D\ P$. where D = diameter of cylinder in inches and P = boiler pressure (gauge.)

The formulas by Barr seem to give the most universal satisfaction.

Crank Pin:—The crank pin should be designed, first, to avoid heating; second, for strength; third, for rigidity. As a matter of fact the first of these factors is the most important one and is generally considered first. Experience has shown that the heat dissipated is proportional to the projected area of the pin and hence the equation $C\ d\ l = P\ A$. or

$$d\ l = \frac{P\ A}{C}$$

where C = pounds pressure per square inch projected area allowed, d = diameter of pin in inches, l = length of pin in inches, P = mean effective pressure in pounds per square inch of piston, and A = area of piston in square inches.

Current engine practice (See Trans. A. S. M. E., Vol. 17, page 124) has shown that the projected areas in a large number of engines averaged according to

$$d\ l = .22\ A \qquad (12)$$

which, if the steam is used at 100 pounds gauge (approximately 50 pounds M. E. P.) reduces to C = 227, say 225 pounds average pressure on each square inch of projected crank pin. This figure corresponds to the minimum value of C as given in the same reference and is considered good working conditions for high speed stationary work. It should be noticed however, that the value of C for low speed engines and for locomotive work would be much larger.

After obtaining $d\ l$ from the above equation, l may be substituted from one of the following equations to find d.

Whitham :—$l = \dfrac{.045\ H.\ P.}{L}$ Allowing .05 as a coefficient of friction.

Marks :— $l = \dfrac{.05\ H.\ P.}{L}$ Allowing .05 as a coefficient of friction.

Thurston :— $l = \dfrac{.06\ H.\ P.}{L}$

Unwin :— $l = \dfrac{.3\ H.\ P.}{r}$

where L = length of stroke in feet, and r = radius of crank in inches.

The following formula is suggested as giving probably the best results. If L' = stroke in inches

$$l = \frac{.3\ H.\ P.}{L'} + 2.5 \qquad (13)$$

$$\left.\begin{array}{ll} \text{(M. A.)} & d = \dfrac{1.265\ P\ D}{N} \\ \text{(M. A.)} & l = .0032\ d\ N \end{array}\right\} \quad \begin{array}{l} \text{where } P = \text{gauge pressure.} \\ D = \text{diameter of cylinder in inches.} \\ d = \text{diameter of pin in inches.} \\ N = \text{R. P. M.} \end{array}$$

These formulas may be used for checking.

It is not an uncommon practice in some work of this class to make the diameter of the pin equal the diameter of the shaft.

Wrist Pin:—The projected area of the wrist pin (Trans. A. S. M. E.) was found to be, from the mean pressure exerted,

$$d\, l = .105\, A \tag{14}$$

Combining this with $$d\, l = \frac{P\, A}{C}$$

gives $C = 476$, say 475 pounds average pressure per square inch of projected area.

Obtain $d\, l$ as before and assume either d or l. It is common to take l for the wrist pin about one inch less than that of the crank pin; or a value for the diameter may be obtained from

$$d = (.9 \text{ to } 1)\, l$$

(M. A.) :—$d = .00325\, P\, D$. where P = gauge pressure.
D = diameter of cylinder in inches.
(M. A.) :—$l = 1.0495\; d$ d = diameter of pin in inches.

Cranks:—Side cranks are usually built up as shown by Fig. 5. The cast iron disc should be fitted to the shaft by a pressed fit. Some forms of crank pins are also fitted in this way.

Allowance for a pressed fit can be taken at .0025″ per inch of diameter of shaft.

For reference on forced fits see:—American Machinist: Feb. 16, '99; May 11, '99; Aug. 10, '99.

For calculations of the simple side crank see formulas under center cranks.

Center Cranks are usually of the solid forged type as shown in Fig. 6.

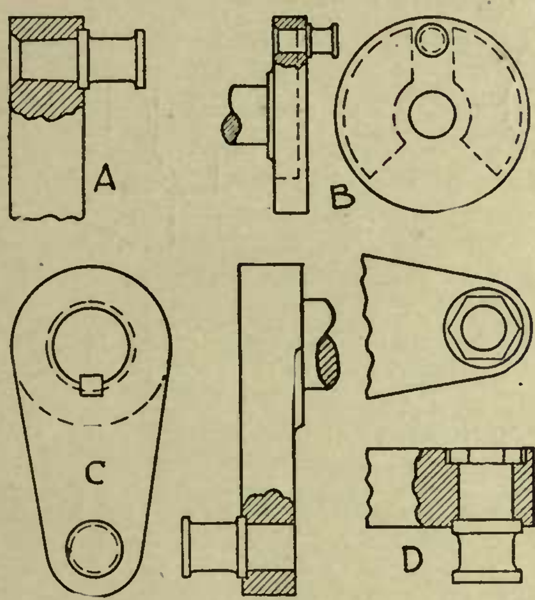


FIG. 5.

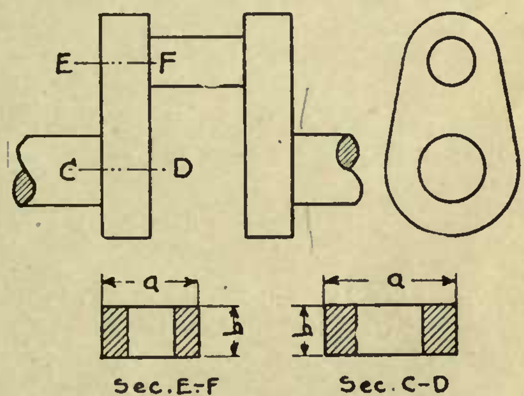


FIG. 6.

To determine the section E. F. put the engine on the head end dead center and assume 100#□″ on the piston as the greatest unbalanced load likely to be used. Then $N = P\, A = (100\, A)$ = force acting. The unit compressional stress on $E\, F$ due to N is

$$f_c = \frac{P\, A}{2\, a'\, b}$$

The unit bending stress due to N is

$$f_b = \frac{P\, A\, M}{\left(\frac{a'\, b^2}{6}\right)2}$$

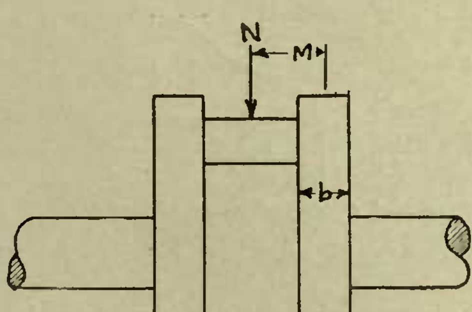


FIG. 7.

Then the total unit stress due to N is

$$f = f_c + f_b = \frac{P\,A}{2}\left(\frac{1}{a'b} + \frac{6\,M}{a'\,b^2}\right) \quad \text{and}$$

$$a' = \frac{P\,A}{2\,f}\left(\frac{1}{b} + \frac{6\,M}{b^2}\right) \tag{15}$$

For a single overhung crank (15) becomes

$$a' = \frac{P\,A}{f}\left(\frac{1}{b} + \frac{6\,M}{b^2}\right) \tag{16}$$

Since there are here two unknowns, a' and b, it will be necessary to assume one to obtain the other. Let $b = \frac{D}{2} + 1''$ where D = diameter of shaft or, (M. A.) :—$b = .45\,D + .5''$. Substitute b in equation (1) and find a'.

In side crank engines, the crank disc is usually of cast iron, and consequently in applying (16), b is made proportionally greater than for center crank engines.

To determine sec. C. D. put the engine so that the crank and connecting rod are at right angles, Fig. 8 and assume that steam follows the piston for half stroke.

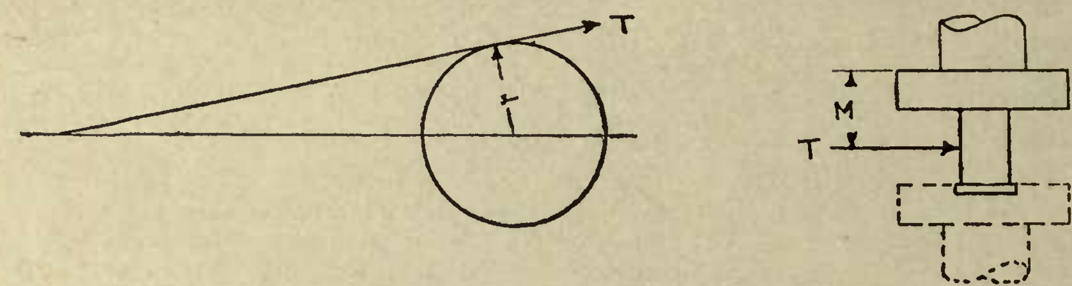


FIG. 8.

We then have a combined bending moment Tr and a twisting moment Tm which according to Unwin, Part I Paragraphs 44 and 126, give an equivalent bending moment of approximately

$$M' = \left(0.91\ Tr + \frac{0.41\ Tm}{2}\right)$$

The bending is parallel to the plane of rotation hence the resistance of the section is $a^2\,b \div 6$ and the formula for dimension of section becomes

$$a = \sqrt{\frac{6\,T}{b\,f}\,(0.91r + 0.205\,m)} \tag{17}$$

This formula becomes for single or overhung cranks.

$$a = \sqrt{\frac{6\,T}{b\,f}\,(0.91r + 0.41\,m)} \tag{18}$$

The rotative effect T on the crank pin, of the horizontal pressure at the crosshead, is

$$T = P\,A\begin{cases} 1.03 \text{ where } r \div l = \frac{1}{4} \\ 1.02 \text{ where } r \div l = \frac{1}{5} \\ 1.01 \text{ where } r \div l = \frac{1}{6} \end{cases}$$

The value of f should be between 1500 for small engines and 2500 for large engines.

Substitute T, f, r, and m and find a. Check this with (M. A.) :— $a = 1.185\,D$ where D = diameter of shaft.

Counter Balance:—In designing the crank we are confronted with the problem of providing a counter-balance for both the revolving and the reciprocating parts. This is a problem, the solution of

which, at best, can be but a compromise. It is an easy matter to balance the rotating parts but a complete balance for the reciprocating parts can not be made. (See St. Eng. and Boiler Notes.)

One of the most satisfactory rules for counter-balancing horizontal engines is given by Unwin

$$W_1 = (\tfrac{2}{3} \text{ to } \tfrac{3}{4})\ (W_2 + W_3)\frac{r}{p}$$

where W_1 = weight of counter-balance in pounds.
p = radius of center of gravity of W_1
W_2 = weight of crank pin and ½ connecting rod in pounds.
W_3 = weight of reciprocating parts in pounds.
r = radius of crank in inches.

Main Bearing:—No very close figures can be given on the length of the main bearing. A rough estimate would be, say, two to three times the shaft diameter. A more satisfactory figure, however, may be obtained from Barr. (Trans. A. S. M. E.). Let $d\ l$ = projected area of bearing, then $d\ l = C\ A$
C = constant varying between .367 and .739
A = area of piston in square inches.
Taking the mean value of C as .489 the formula becomes—

$$d\ l = .489\ A$$

With the mean value of C and a steam pressure of 100 pounds gauge, the average pressure per square inch of projected area of the bearing becomes approximately 100 pounds. If the minimum and maximum values of C were used, the corresponding pressures would approximate 140 and 70 pounds respectively.

The average size taken from a number of side crank engines seems to show

$$d\ l = .7\ A$$

while for center crank engines with two bearings of equal size the value approximates

$$d\ l = .55\ A$$

from which, knowing the shaft diameter, the length of the bearing can be obtained.

Connecting Rod:—In designing the connecting rod it should be considered, first, as a rod in direct tension and compression, in which case a fibre stress of not more than 3000 pounds per square inch should be used. It should then be treated as a long column which can buckle in two ways; first, in the plane in which it moves, in which case it is pin ended; second, in the plane perpendicular to the plane in which it moves in which case it may be regarded as flat ended. Apply Rankine's or Euler's formula for columns and determine if the calculated section is safe. The first formula will more satisfactorily apply in this case. See Church Par. 307.

Ratio of breadth to height of rod section.—In Euler's formula P = force which will produce incipient flexure, I' and I'' are moments of inertia of the sections; then considering a rectangular section, if b = breadth and h = the height of the section, $I' = \frac{1}{12}\ b\ h^3$ and $I'' = \frac{1}{12}\ b^3\ h$. If we make the rod equally strong in both directions we have $P' = P''$ or

$$\frac{E\ I'\ \pi^2}{l^2} = \frac{4\ E\ I''\ \pi^2}{l^2} \text{ or } I' = 4\ I''$$

then $\frac{1}{12}\ b\ h^3 = \frac{1}{3}\ b^3\ h$ or $h = 2\ b$. Hence, we learn that the breadth should be one-half the height of the section to be uniformly strong in each plane. The above is based on the assumption that the column is uniform throughout and that the centrifugal force has no effect on the rod. As a matter of fact, in practice the section varies; being smallest at the wrist pin. The rod is also subjected to a whipping action, such that, on a high speed engine this factor has considerable effect. No very definite information on this point can be obtained, but it can be proven that with a rod of uniform section throughout, the greatest strain comes at a point about .6 the length of the rod from the wrist pin. See following problem.

Because of the fact that connecting rods are subjected to such severe and uncertain forces, manufacturers make the height of the sections $h = (2b$ to $3b)$ the average value being $h = 2.5b$. It will also be noticed that rectangular rods increase in section as they approach the crank pin. The amount of taper is probably more a factor of neatness in design than of required strength. The rod should be figured for strength and rigidity and use this section next to the wrist pin, whatever excess is then added for taper will simply increase the factor of safety.

Kent gives as the distance between the parallel sides of a rectangular rod

$$T = .01\ D\ \sqrt{P} + .6''$$

where D = diameter of cylinder in inches and P = max. unbalanced pressure in pounds per square inch.

Rods having rectangular section are generally used on high speed engines. If reference to sizes of rods having other shaped sections is desired, look up Kent, page 799.

The following sections are used on connecting rods.

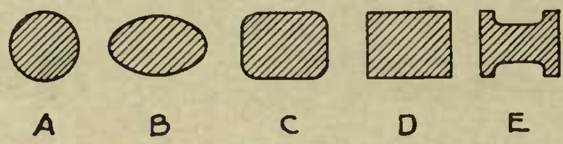


Fig. 9.

A. Slow speed. Corliss Type. Large at center, smaller at each end.
B. Elliptical. Used on straight line engines.
C. High Speed Engine Type.
D. High Speed Engine Type.
E. Locomotive Type.

For shapes and sizes of rod ends and brasses see Unwin, Vol. II, Pages 108-120; Low & Bevis, pages 205-212; "The Constructor," pages 112-113; Whitham, pages 217-227; Marks, pages 53-61; Kent, page 800.

Greatest Bending Moment in the Connecting Rod:—*Problem*—Find the point in the connecting rod Fig. 10, that is subject to the greatest bending moment. $w v^2 \div g r =$ Centrifugal load at the pin where $w =$ weight in pounds per unit of length. With the load at the cross head = zero, we have the conditions shown in Fig. 11, i. e., a beam $A B$ sustaining a load varying uniformly from $w v^2 \div g r$ to 0. Assume a section at $a b$, a distance x from the supports, and we have

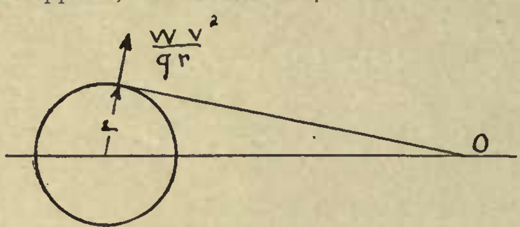


Fig. 10.

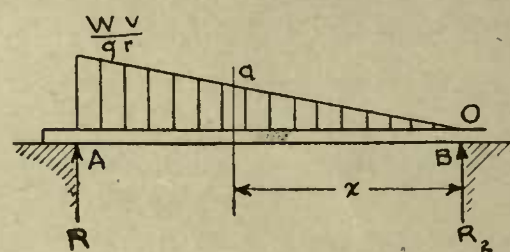


Fig. 11.

$$R_1 l = \int_0^l \frac{w v^2}{g r} \frac{x}{l} x \, dx = \left[\frac{w v^2}{g r} \frac{x^3}{3 l} \right]_0^l = \frac{w v^2 l^3}{3 g r l} \quad \therefore \quad R_1 = \frac{w v^2 l}{3 g r}$$

$$\text{also} \quad R_2 l = \int_l^0 \frac{w v^2}{g r} \frac{l - x}{l} x \, dx = \left[\frac{w v^2}{g r} \left(\frac{x^2}{2} - \frac{x^3}{3 l} \right) \right]_l^0 = \frac{w v^2 l^2}{6 g r} \quad \therefore \quad R_2 = \frac{w v^2 l}{6 g r}$$

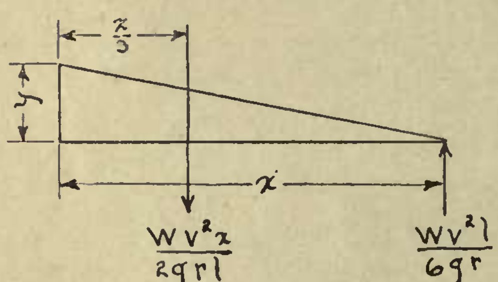


Fig. 12.

To find the *greatest bending moment*, consider the section x as free, and Fig. 12.

$$M = \frac{w v^2 l x}{6 g r} - \frac{w v^2 x^3}{6 g r l}$$

$$\frac{d M}{d x} = \frac{2 l}{6} - \frac{x^2}{l} = 0$$

$$\therefore 2 l^2 = 6 x^2$$

$$x = \frac{l}{\sqrt{3}} = .6 l$$

Another way to find the greatest bending moment is to equate the shear to zero.

$$F = \frac{w v^2 l}{6 g r} - \frac{w v^2 x^2}{2 g r l} = 0 \quad \therefore \quad \frac{l}{6} = \frac{x^2}{2 l}, \qquad 2 l^2 = 6 x^2$$

$$x = \frac{l}{\sqrt{3}} = 6 l$$

Cross Head:—The maximum pressure on the cross head will be, with cut-off at half stroke, $W = P\ A\ \tan\ \Phi$ where P = maximum unbalanced pressure on piston in pounds per square inch; but $\tan\ \Phi = r \div l$ and $W = P\ A\ (r \div l)$. Also, $W = w\ B$, where w = pressure in pounds per square inch of bearing surface and B = projected surface of one side of the cross head in square inches, then

$$w\ B = P\ A\ \frac{r}{l}$$

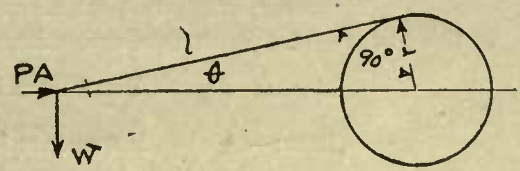


FIG. 13.

The value of w is taken as follows: Rankine, 72.2; Whitham, 100; "The Constructor, 60 to 120; Thurston, 66 to 100.

There seems to be a wide diversity of practice in fixing the values of w, some builders requiring a value as low as 30 while others will permit as much as ten times that amount. It is desirable in high speed engine work to have large surfaces and low pressures. If $w = 30$, $P = 100$, and $r \div l = 6$, then $B = .55\ A$.

For mean values Barr gives $B = .63\ A$ for high speed engines, and $.46\ A$ for low speed engines.

(M. A.) $\quad B = .75$ H. P.

It may not be possible on account of some feature of the design to keep to these figures, but the object should be, to assume B such that w will be between 30 and 75 pounds per square inch.

For *cylindrical cross heads,* the *diameter* may be taken one inch larger than the diameter of the cylinder.

The *length* of the average cross head is $l = .9\ L$ where L = length of stroke in inches.

For details of cross heads, look up:—

Unwin	Vol. II, pages 121-136
Low & Bevis	pages 218-228
"The Constructor"	Pages 118-121
Whitham	pages 208-216
Marks	pages 44- 53
Kent	page 798

Piston Rod:—The piston rod may be regarded first as a piece in direct tension and compression, in which case the allowable stress would be 3,000 to 5,000 pounds per square inch, calculated at the weakest section. Again, it should be considered as a long column under compression, and by the use of Euler's or Rankine's formula for pin and square ends, determine if there is any danger of buckling when under compression. The length of the unsupported part of the rod is about $1.25\ L$.

The following formulas may be found useful in determining the diameter of the rod.

Barr:— $\begin{cases} d = .11\ \sqrt{D\ L} & \text{for high speed engines.} \\ d = .145\ \sqrt{D\ L} & \text{for low speed engines.} \end{cases}$

Kent:— $d = .013\ \sqrt{D\ L\ P}$

Whitham:— $d = K\ D$ where $K = .16$ for steel.

Seaton:— $d = \frac{D}{F}\ \sqrt{P}$ where $F = 45$ to 50.

Marks:— $d = .03525\ \sqrt[4]{D^2\ L^2\ p}$

Unwin:— $d = .0144\ D\ \sqrt{P}$

Thurston:— $d = \sqrt[4]{\frac{D^2\ p\ L^2}{a}} + .0125\ D.$ $\begin{cases} \text{where } a = 10000 \text{ for H. S. Engines.} \\ a = 15000 \text{ for L. S. Engines.} \\ \text{and } L = \text{length of stroke in ft.} \end{cases}$

Low & Bevis:— $d = .018\ D\ \sqrt{P}$

(M. A.):— $d = .018\ D\ \sqrt{P}$

In all the above the following notation prevails, except where otherwise stated,

L = length of stroke in inches.
D = diameter of cylinder in inches.
P = maximum unbalanced pressure in pounds per square inch.

Piston Face:—The piston face varies from $.2\ D$ to $.5\ D$.

Marks:— Piston face $l = \sqrt[4]{L\ D.}$

Barr:— Piston face $l = \begin{cases} .46\ D \text{ for high speed engines.} \\ .32\ D \text{ for low speed engines.} \end{cases}$

I. C. S.:— Piston face $l = .2\ D + 1.5''$

(M. A.):—Piston face $l = .43\ D$.

In all the above, L = length of cylinder in inches and D = diameter of cylinder in inches.
For shapes of piston details, see

Whitham .... Pages 11- 25
Unwin .... Vol. II, Pages 136-159
Low & Bevis .... Pages 229-238
Holmes .... Pages 208-213

Packing Rings:—Packing rings are turned about $\frac{3}{16}''$, to each foot of diameter, larger than the cylinder, and a portion is then cut out of them so the ends just come together when fitted for use. A good thickness for rings is

$$t = .03\ D$$

(M. A.): — $t = .8\ w$ where w = width of ring in inches.

The width of rings varies a great deal according to various authorities.

Unwin $w = .014\ D + .08''$.

Whitham $w = .15\ D$

In the two formulas the width of the ring would vary in a 10″ cylinder by a factor of 7. In practice, the piston ring varies from a width of ⅜″ in the smaller sizes to ¾″ in the larger sizes.

(M. A.):— $w = .055\ D$.

Cylinder Walls:—It will be found, in figuring the cylinder walls to resist the internal pressure, that even if we allow a large factor of safety, the calculated sizes run much below the sizes commonly used in practice. The cylinder is designed heavy for other reasons, chiefly for rigidity and for reboring. Probably the most commonly used formula for the cylinder thickness is

Barr:— $t = .05\ D + .3''$

where D = diameter of cylinder in inches.

t = thickness of walls in inches.

This formula agrees with practice in smaller sized engines but gives results below practice in the larger engines. The following may be considered very close to the average.

(M. A.):— $t = .07\ D + .125''$

The above formulas apply to a steam pressure of 100 pounds. If the steam pressure is increased the constant should also be increased.

Cylinder Flanges:—The thickness of the cylinder flange is usually taken at a certain per cent of the cylinder thickness.

Barr:— $t' = 1.25\ t$

where t = cylinder wall thickness.

(M. A.):— $t' = .12\ D$.

where D = diameter of cylinder in inches.

Cylinder Head:—The cylinder head will be from 1 to 1.5 times the thickness of the cylinder. It should fit to the cylinder metal to metal. Ordinarily there will be two cylinder heads considered;

head end, and crank end. The crank end head forms the dividing line between the bed and the cylinder. There are four methods of construction here.

(1) Bed and cylinder cast solid, used only on very small engines.
(2) Cylinder head separate and bolted between the cylinder and bed.
(3) Cylinder head part of the cylinder.
(4) Cylinder head part of the bed.

Cylinder Head Studs:—The number of studs may be taken

$$N = 0.7\,D$$

Always use an even number of studs. The diameter is usually not less than 5/8″ and varies from this to say 1 3/8″. The diameter should be figured from the tension on the area at the root of the thread. This may be then compared with

Barr:— $d = .025\,D + .5''$

Counter Bore:—The counter bores should be such that the packing rings will pass the entire length of the cylinder. Allowance of 1/4″ to 1/2″ in diameter is made. The counter bore should *curve* down to meet the cylinder proper.

All drips are tapped into the counter bore.

Steam Pipe:—The velocity of the steam for engine service will vary from 5000 to 6000 say 5500 F. P. M. A rational formula for the amount of steam passing through a pipe is

$$Q = \frac{C\,a}{144} \tag{19}$$

where Q = volume of steam in cubic feet. C = velocity in F. P. M. a = area of pipe in square inches.

The formula for the steam engine cylinder is

$$Q = \frac{A\,V}{144} \tag{20}$$

where A = area of piston in square inches, and V = velocity of piston in F. P. M.

Equating we have $\frac{C\,a}{144} = \frac{A\,V}{144}$ or $\frac{A\,V}{C} = a$

Now, if $V = 600$ and $C = 5500$ then, $a = .11\,A$. Check this with

Kent:— $d = .408\,\sqrt{H.\,P.}$

Exhaust Pipe:—Low pressure steam will not flow as fast as high pressure steam, hence, C should be taken less, (2500 to 5000, Barr), say 3800 then as above

$$a = .16\,A$$

Kent:— a = 25 to 50 per cent greater than steam pipe.

Cylinder Ports:—The ports or passages, through which the steam enters the cylinder, should be short, direct, of easy curvature and large enough to prevent wire drawing of the steam. It seems to be current practice to figure the area of the port from a steam velocity of from 5000 to 5500 F. P. M. Assuming the lower figure for ports that handle both live and exhaust steam, we have

$$a = .12\,A.$$

Other formulas for area of ports.

Hubbard	$a = .1\,A$
I. C. S.	$a = .11\,A$
(M. A.)	$a = .113\,A$

The length of the port varies from $.7\,D$ to D.

I. C. S.	$l = .7\,D$
Hubbard	$l = .75\,D$
(M. A.)	$l = .95\,D$

Recent practice is making the length of the port about equal to the diameter of the cylinder.

To serve as a check in working up the Zeuner, the following ratios are added. These must not be considered as fixed values, they merely represent the average, as others before given.

Valve Travel (Max.) $S = .269\,L$.

Steam Lap = width of port.

STEAM BOILER DESIGN.

General.

In designing a steam boiler the following points should be kept in mind.

First, the boiler should have a good circulation of the water and should steam rapidly. To have a good circulation, the water must have a free path from the point of greatest heat to all points within the boiler and this path must not be seriously obstructed by short or abrupt turns. To be a good steaming boiler it must have a large amount of heating surface, a large grate area and a water space well broken up into small volumes. The term *heating surface* is understood to refer to any surface having the heated gases on one side and water on the other. No surface in the steam space should be exposed to the heated gases.

Second, the boiler should be built compactly and should be easily accessible for cleaning and repairs. Floor space in the average boiler room is valuable and other things being equal the boiler requiring the least floor space will be first considered by the purchaser. The cleaning of a boiler is also an important matter in power house work and the designer should so arrange the parts of the boiler that all may be easily reached.

Third, the boiler should be strong enough to be safe under its rated steam pressure and should have provision for the expansion and contraction of its parts without endangering those parts, or the boiler setting which surrounds them.

There are two general classes of boilers, *fire tube* and *water tube*. The fire tube boiler is the old form and is built vertical or horizontal, tubular or flue. The typical representative fire tube boiler is the horizontal tubular or "Multitubular" boiler as it is called. The water tube boiler is of more modern design, and is usually built in sections. Typical forms of this class of boilers are the Babcock and Wilcox, Sterling, Wickes and Heine. The chief features which distinguish the water tube from the fire tube boiler are, first, water circulation on the inside of the tubes; second, reduced water space and its subdivision into smaller units; third, increased heating surface.

PROBLEMS IN MULTITUBULAR BOILER DESIGN.

Having given the steam consumption of the engine in the previous design, find the equivalent evaporation of the boiler. It would be well here to assume a steam consumption of the engine about 25 per cent greater than the *rated* amount.. The boiler, under such conditions, would then supply enough steam to allow for a certain overload on the engine.

To obtain this equivalent evaporation take—(Engine horse power) $\times$ (pounds of steam per horse power hour) $\times$ (total heat in steam at absolute pressure—total heat in water entering boiler at, say, 212 degrees F.) divided by (34.5×965.6)

PROBLEM 1. Determine the following dimensions for a . . . . H. P. boiler. (a) Heating surface; (b) Grate surface; (c) Diameter and length of tubes; (d) Heating surface in tubes. ($\frac{8}{9}$ of the total H. S. approx.) ; (e) Number of tubes; (f) Ratio of the total tube area to grate area. For size of boiler tube area see Kent, page 196; (g) Volume of the shell. (water space + steam space + volume of tubes); (h) Diameter of the boiler shell; (i) Height of water line.

PROBLEM 2. Lay out the tube sheet. This should be done experimentally on a trial sheet.

PROBLEM 3. After completing the lay-out of the tube sheet, check the calculations in Problem 1 on the following points: heating surface, nominal rate of evaporation, ratio of the tube openings to grate area, and volume of shell.

PROBLEM 4. Determine the distance E; Fig. 14, also the total stack area.

PROBLEM 5. Find the area above the tubes in the tube sheet and determine the total load on the stays.

PROBLEM 6. Locate the boiler stays.

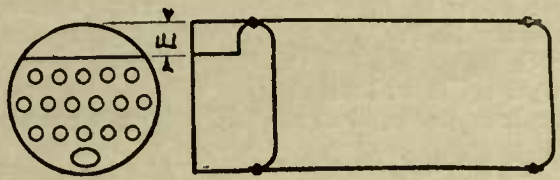


FIG. 14.

PROBLEM 7. Determine the size and the spacing of the rivets to prevent the failure of the stay bolt fastening.

PROBLEM 8. Determine the size of the stay bolt body.

PROBLEM 9. Plan the longitudinal seams of the boiler shell and calculate the efficiency of the same.

PROBLEM 10. Plan the girth seams and calculate the efficiency of the same.

PROBLEM 11. Calculate the factor of safety in the entire boiler taking into account the strength of the joints.

PROBLEM 12. Determine the diameter of the steam pipe, feed pipe and blow-off pipe.

PROBLEM 13. Estimate the total weight of the boiler.

PROBLEM 14. Estimate the weight of the boiler, when working under normal conditions.

(a)	Weight of boiler and accessories,	(..) pounds.
(b)	Weight of water in boiler	(..) pounds.
(c)	Weight of steam in boiler.	(..) pounds.
Grand Total,		(..) pounds.

PROBLEM 15. Calculate the number of rivets in each boiler leg or support.

PROBLEM 16. Calculate the intensity of the pressure between the leg and the boiler setting.

NOTES ON THE DESIGN OF MULTITUBULAR BOILERS.

References:—The designer is expected to consult Hutton's "Mechanical Engineering of Power Plants," Peabody and Miller's "Steam Boilers," Whitham's "Constructive Steam Engineering," "Kent," trade catalogues and the like and become familiar with the general shapes in the design of boiler parts. He should also study the average boiler setting and note the methods of construction in furnace, grates, bridge wall, flame way, rear arch, front connections, smoke nozzle and breeching. Two plates of drawings accompany these notes. They represent average student work and are added merely to illustrate how the sheets may be compiled. It is requested that each designer make a report containing the theoretical part of the work and that only one complete set of plates need be made as a part of the engine set. These tracings may be retained at the University but the reports will be returned at graduation. The horse power of the boiler to be designed will be such as to supply steam to the engine just completed.

Capacity of the Boiler:—This will depend upon three things; grate area, heating surface, and steam and water space. Of the three, the heating surface is of prime importance. A very satisfactory definition of heating surface is found in Kent, page 679. Heating surface $=$ ⅔ area of shell $+$ area of tubes $+$ ⅔ area both heads $-$ 2 $\times$ cross sectional area of all tubes. The heating surface of tubes should be measured on the inside.

Commercial Boiler Rating:—Boilers are usually rated in the term horse power. They are designed on the basis of the steam consumption of the engines they are to supply. Boilers are installed however, of larger capacity than that indicated by the commercial engine rating because of the usual under-rating of the engines and because it is desired to supply ample steam without forcing. Twenty-five per cent increase would represent a fair average. A boiler horse power is stated as follows:

Horse Power:—One boiler horse power equals 30 pounds of steam evaporated from feed water at 100° F. to dry steam at 70 pounds gauge pressure, or 34.5 pounds of water evaporated from and at 212° F.

All engines do not require the same amount of steam per horse power. The following values may be taken for high speed engine practice. Simple, 30 to 32 pounds per H. P. hour. Compound non-condensing, 24 to 26 pounds per H. P. hour. Compound condensing, 18 to 20 pounds per H. P. hour.

Heating Surface and Grate Surface:—

Heating surface per Boiler H. P.	11.5	square feet.
Grate surface per Boiler H. P.	.33	square feet.
Ratio of H. S. to G. S.	34.5	
Water evaporated per sq. ft. of heating surface per hour from and at 212° F.	3	pounds.

The grate surface is from 30 to 50 per cent air space.

Tubes:—The tubes must be of sufficient area to carry away the products of combustion. A good proportion is

Area of tubes : Area of the grate = 1 : 8

The length of the tubes is about 5 feet for each one inch in diameter. Outside diameter of tubes = 2″, 2½″, 3″, 3½″ and 4″. In general, large boilers have large tubes and small boilers have small tubes; say 20 H. P. 2″; 60 H. P. 2½″; 100 H. P. 3″, etc. These values are not definitely fixed but are a fair average. From $\frac{7}{8}$ to $\frac{8}{9}$ of all the heating surface is in the tubes.

Steam Space:—Allow .8 to 1.0 cubic foot per horse power. This amounts to approximately ¼ the interior volume of the boiler.

Length of the Boiler:—The length of the boiler proper is the overall length at the tube sheets. Having found this the designer then selects between flush front and extended front as shown in Figs. 15 and 16. The length x of the smoke box can be taken 0.08 H. P. + 10″. The Atlas Engine Company of Indianapolis makes x constant for all sizes = 15″.

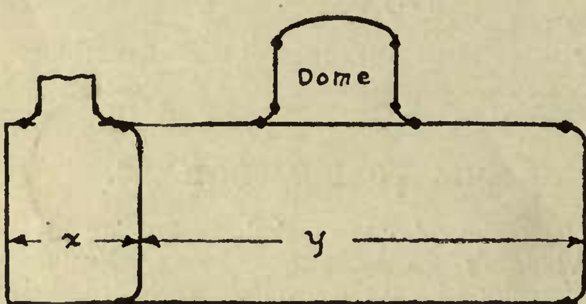


FIG. 15.

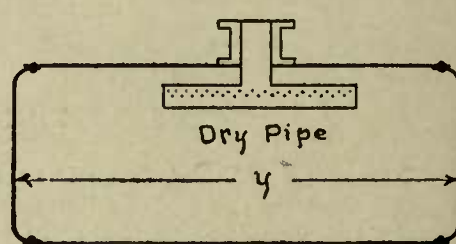


FIG. 16.

Stack Area and Area Over Bridge Wall:—The following relation holds good for area of stack and area over bridge wall.

Area of stack : Area of Grate = 1 : 9
Area over bridge wall : Area of Grate = 1 : 7

Materials. General:—The materials used in boiler work are:

Shell; Mild Steel: fire-box, flange and open hearth.
Rivets and *Braces;* wrought iron preferred.
Fittings; cast iron, malleable iron and mild steel.

On all plates, the elastic limit should be at least one half the ultimate strength. The percentage of manganese and carbon are left to the judgment of the steel maker.

Steel up to ½ inch thickness must stand bending double and being hammered down on itself, above that thickness it must bend round a mandrel having a diameter of 1½ times the thickness of the plate, down to 180°, all without showing signs of distress.

Test pieces are cut both lengthwise and crosswise of the plate, each piece to have a length not less than 16 × thickness of plate. All rough sheared edges should be milled or filed off.

Important Qualities:—

Shell plates not exposed to direct fire.	Tensile strength, 65000, 70000#□″ Elongation—Not less than 24% in 8″ Phosphorus—Not over .035% Sulphur—Not over .035%
Shell plates exposed to direct fire, or plates on which flanging is to be done.	Tensile strength, 60000, 65000#□″ Elongation not less than 27% in 8″ Phosphorus not over .03% Sulphur not over .025%
Fire box plates, exposed to direct fire or flanged over the greater portion of their periphery.	Tensile strength 55000-62000#□″ Elongation 30% in 8″ Phosphorus not over .03% Sulphur not over. .025%

Cast Iron:—All cast iron shall have a soft, gray texture and a high degreee of ductility. It will be used only on crabs, yokes, hand hole plates, man heads, etc. It should not be used on mud drums, legs, necks, headers, manhole rings or any part of the boiler subjected to tensile stress.

Shell:—Boiler shells until recently, were wrought iron, but with the advent of high steam pressures, open hearth, or crucible steel containing about ¼ of one per cent of carbon replaced it. The ultimate strength of this steel is 55000 to 70000#□″. Allowable fiber stress about 8000#□″. Steel plate is more susceptible to injury during forming than wrought iron plate.

The number of riveted seams in a boiler should be as few as possible. In small boilers it is possible to make one longitudinal seam suffice, having the length of the plate running lengthwise with the boiler. Fig. 17 A. In large boilers the plates become too large to be cheaply produced and two or more sheets are used as in B. Generally however, boilers are made from rings as in Fig. 18.

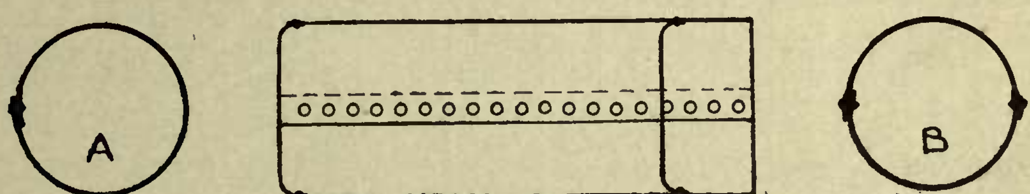


Fig. 17.

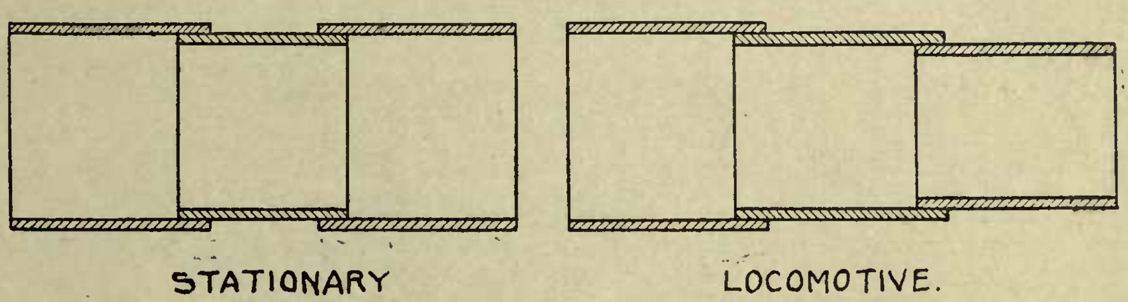
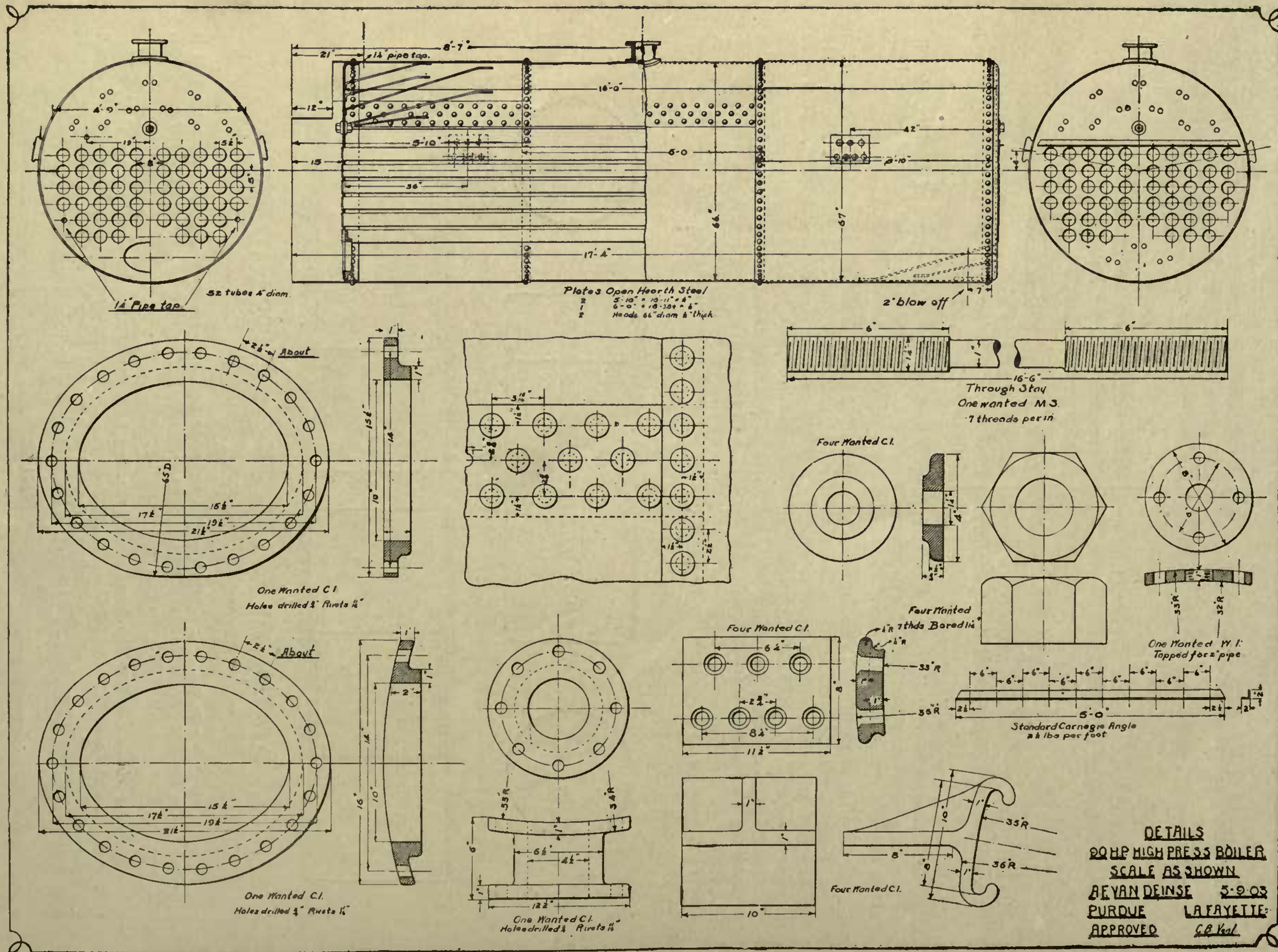


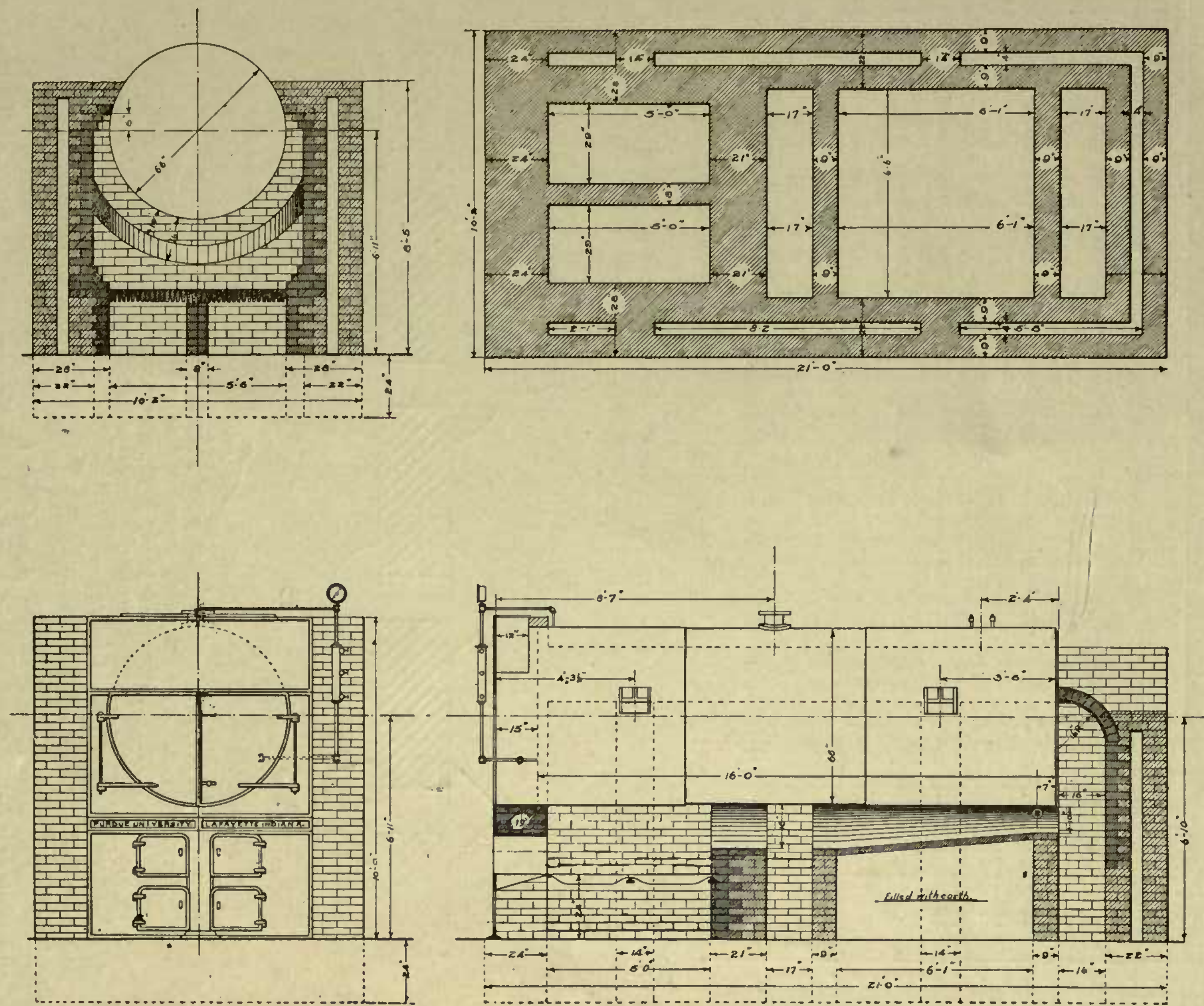
Fig. 18.

When a sheet is selected, it is marked with a template and either punched or drilled, after this it is rolled to the desired curvature and riveted. The holes are prepared for riveting by either punching, which is the cheapest method, punching and reaming, or drilling.

Thickness of the Shells—The force tending to burst the boiler shell longitudinally is $P\ D$, where P = gauge pressure in pounds and D = diameter of shell in inches. The resisting force is $2\ t\ f$, hence the thickness of plate $= P\ D \div 2\ f$. Since the thickness of the plate to resist transverse rupture is only half as great, ($t = P\ D \div 4\ f$) the first formula should be used for boiler shells.



1½" Pipe tap
52 tubes 4" diam.
1½" pipe tap.
Plates Open Hearth Steel
Heads 66" diam ½" thick
2" blow off
Through Stay
One wanted M.S.
7 threads per in
Four Wanted C.I.
Four Wanted
7 thds Bored 1½"
One Wanted W.I.
Topped for 2" pipe
Standard Carnegie Angle
One Wanted C.I.
Holes drilled ¾" Rivets ¾"
Four Wanted C.I.
Standard Carnegie Angle
One Wanted C.I.
Holes drilled ¾" Rivets 1½"
One Wanted C.I.
Holes drilled ¾" Rivets ¾"
Four Wanted C.I.
DETAILS
20 HP HIGH PRESS BOILER
SCALE AS SHOWN
A E VAN DEINSE 5-9-03
PURDUE LAFAYETTE
APPROVED G.B. Veal



SETTING PLANS
90 H.P. HIGH PRESS BOILER
SCALE ½"-1'

A.E. VAN DEINSE 3-20-03
PURDUE LAFAYETTE
APPROVED

Kent gives $t = \dfrac{P\,D}{2\,f\,c}$ where $f = 10000$#□″ and $c = .70$ for a double riveted longitudinal seam.

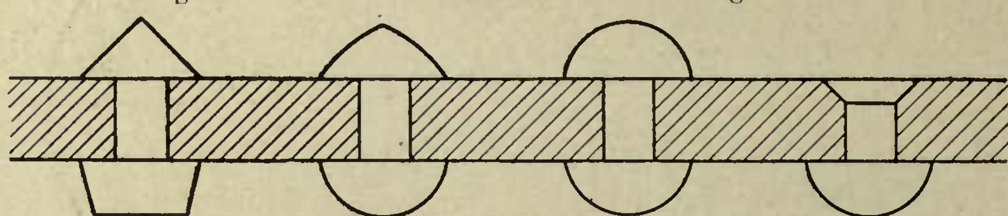
The following data shows the maximum sizes of plates that may be ordered from the rolling mills. (1904).

LARGE MILL.---MAXIMUM SIZES OF PLATES.

Thickness of Plates	Width of Plate in Inches															Diam. of head that can be rolled
	144	140	135	130	125	120	110	100	90	80	70	60	50	40	30	
¼″							150	180	240	260	280	300	340	360	420	124
5/16″					220	240	250	260	270	280	300	340	360	390	510	140
⅜″		180	200	220	240	280	300	340	360	380	400	420	480	520	560	144
7/16″		180	200	220	240	300	340	380	420	440	480	520	540	560	600	144
½″	144	200	220	240	260	300	340	380	420	440	480	520	540	560	600	146
9/16″	144	200	220	240	260	300	340	380	420	440	480	520	540	560	600	146
⅝″	144	200	220	240	260	300	340	380	420	440	480	520	540	560	600	146
11/16″	144	200	220	240	260	300	340	380	420	440	480	520	540	560	600	146
¾″	144	200	220	240	260	300	340	380	400	440	480	520	540	560	600	146
13/16″	144	200	220	240	260	300	340	380	400	420	460	480	500	540	580	146
⅞″	144	200	220	240	260	300	340	380	400	420	460	480	500	540	580	146
15/16″		200	220	240	260	300	340	380	400	420	460	480	500	540	560	146
1″		180	220	240	260	300	340	380	400	420	460	480	500	540	560	146
1⅛″		180	210	240	260	280	320	360	380	400	440	460	480	500	540	144
1¼″		180	210	240	260	280	320	360	380	400	440	460	480	500	540	144
1⅜″		180	210	220	240	260	300	340	360	380	420	440	460	480	500	144
1½″		160	200	220	240	260	300	340	360	380	420	440	460	480	500	144
1⅝″		160	200	210	220	240	260	300	320	340	380	400	400	420	480	142
1¾″		150	190	200	210	220	240	260	280	320	360	380	380	400	460	142
2″		140	170	180	190	200	220	240	260	300	340	360	360	380	440	142
	Length of Plate in Inches															

Rivets:—All rivets shall be good charcoal iron or soft mild steel, having the same properties as fire box plates. They must test hot and cold by drawing down on an anvil with the head in a die. They must also test by nicking and bending and by bending back on themselves cold, all without developing cracks or flaws.

The tensional fibre stress of all wrought iron rivets should not be taken at more than 6000 pounds per square inch. The general forms of rivet heads are shown in Fig. 21.



A B C D

FIG. 21.

A................Conical head, usually hand work.
B................Cup head, swage and snap die work.
C................Rounded head, machine work.
D................Countersunk.

Approximate values of riveted joints:—

The Diameter of rivets for different thickness of plate according to Unwin, Part I, page 103, is, if t = thickness of plate in inches

$$d = 1.2 \sqrt{t}$$

The Pitch of the rivet (minimum) is, $p = 2\,d$. The distance from the center of the rivet to the edge of the plate is, $a = 1.5\,d + 1\text{-}16$ inches.

Tube Sheet:—The thickness of the tube sheet is generally $\frac{1}{16}$ to $\frac{1}{8}$ inch thicker than the shell. The holes for the tubes should be punched ⅛ inch less than the required diameter and reamed to full size, or drilled, and slightly countersunk on both sides. The hole should finish $\frac{1}{64}$ to $\frac{1}{16}$ inch larger than the diameter of the tube, for small and large tubes respectively. Where copper ferrules are used the hole should be a neat fit for the ferrule. Nos. 18 to 14 wire gauge copper should be used in fire tube boilers on ends subjected to the direct heat.

The tube sheet should be annealed after punching and before reaming.

Tube ends in the furnace are beaded to the tube sheet; those in the smoke box are usually rolled to fit and allowed to project ¼ to ⅜ inch beyond the tube sheet.

Bracing the Tube Sheet:—The tubes are supposed to give sufficient rigidity to that part of the boiler head filled by them; the upper segment, however, and occasionally the extreme lower part surrounding the manhole or handhole, needs staying. Boiler stays are of four kinds; *through* stays, which are the simplest and most effective and are used where heavy forces are to be resisted, especially near the central part of the boiler; *angle* stays, used where medium lengths are desired; *gusset* stays, for the short bracing between the head and the shell; and the *girder* stay or *crown bar*, which is used generally to strengthen the crown sheet and upper plates around the fire box.

The *stay* bolt is a form of *through* stay which is used to support the flat sides around the fire box. Fig. 22. shows some of the methods of fastening the various forms.

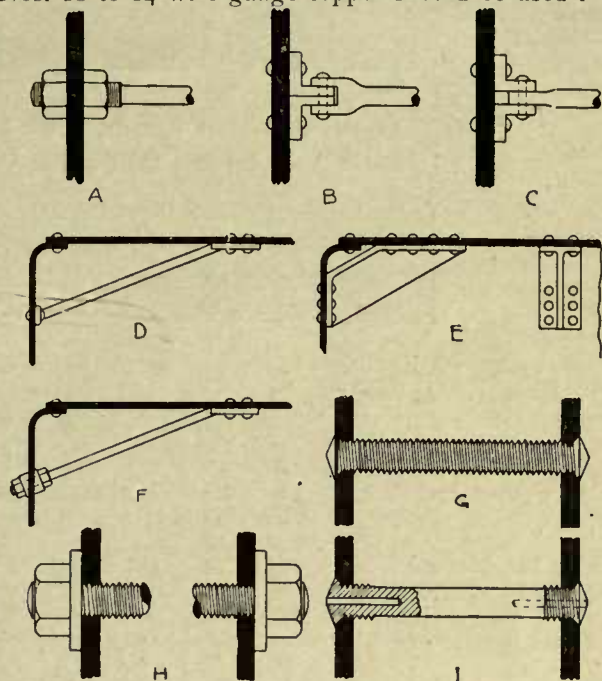


FIG. 22.

To find the *area to be stayed,* let the shell support from 2 to 3 inches and the tubes from 1 to 2 inches above the upper line of the tubes, Fig. 23. Find the remaining area of the segment and, if p is the unbalanced pressure on the head in pounds per square inch, the total load to be stayed will be $p\ A$. Next find the number of stays to be used. The following formula is given for use in spacing the stays.

$$a = t\sqrt{\frac{9f}{2p}}$$

a = distance between stays in inches.
t = thickness of head in inches.
f = Working fibre stress of the metal 6000 for wrought iron and 8000 for special steel.
p = boiler pressure in pounds per square in.

Take this value as the approximate distance between stay centers, and from the load on each stay calculate its size of cross section. Calculate also for a safe fastening to the shell or head.

The load on the head is perpendicular to the tube sheet consequently the actual load on any angle stay will be, Fig. 24.

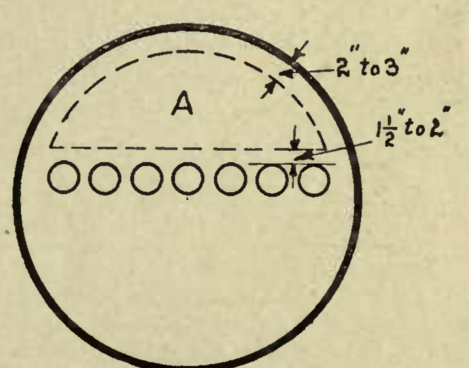


FIG. 23.

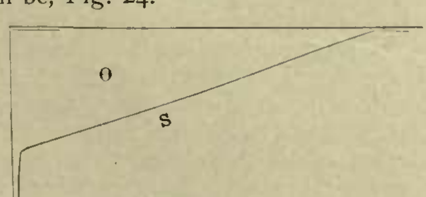


FIG. 24.

$$S = \frac{\text{load to be stayed}}{\cos\theta} \quad \text{or}$$

$$\text{Area of angle stay} = \frac{\text{area of through stay}}{\cos\theta}$$

The angle θ which an angle brace makes with the shell should never exceed 30 degrees, it is preferred that the angle be about 18 or 20 degrees.

Where through stays are used they should be spaced so they will not cut off entrance to the boiler from above.

The material used in stays should be iron or mild steel especially manufactured for the purpose and must show:

Iron	Tensile strength not less than 46000#□″
	Elongation not less than 22% for bolts less than 1□″ area
	Elongation not less than 20% for bolts more than 1□″ area
	Elastic limit not less than 26000 #□″
Steel	Tensile strength not less than 55000#□″
	Elongation not less than 25% for bolts less than 1□″ area
	Elongation not less than 22% for bolts more than 1□″ area
	Elastic limit not less than 33000 #□″

Tests:—A bar taken at random from a lot of 1000 pounds or less, threaded with a sharp die "V" thread with round edges, must bend cold 180 degrees around a bar of same diameter without showing any crack or flaw.

Boiler Tubes:—All tubes shall be charcoal Iron or Mild Steel specially made for the purpose, lap welded and drawn. They must be round, straight, free from scales, blisters and mechanical defects, and each tested to 500 pounds per square inch inter hydrostatic pressure.

Mechanical Test.—A section from one tube taken at random from a lot of 150 or less, must stand hammering down cold vertically without cracking or splitting when down solid.

Length of test pieces.

¾″	for tubes from	1″	to 1¾″	diam.
1″	for tubes from	2″	to 2½″	diam.
1¼″	for tubes from	2¾″	to 3¼″	diam.
1½″	for tubes from	3½″	to 4″	diam.
1¾″	for tubes from	4½″	to 5″	diam.

In arranging the tubes in the tube sheet of the average boiler locate the upper line of tubes so there will be approximately 6 inches between the normal water line and the top of the tubes. Stationary boilers usually have the tubes in vertical and horizontal rows with a large space down the center to aid circulation. Allow from ¾ inch to 1⅛ inch between vertical rows and from ½ inch to ¾ inch between horizontal rows. Locomotive boiler tubes are staggered on 30 degree lines. This gives a possibility of putting in a greater number of tubes than if they were in vertical and horizontal rows. Tubes should not come within 2 or 3 inches of the shell. Allow room for hand holes and manholes in the upper or lower part of the head for cleaning. A manhole is preferred.

Manholes and Handholes:—For shapes and sizes of openings and plates to fit them see "Ryerson's Monthly Journal and Stock List." See also shapes and sizes of crabs.

Dome and Dry Pipes:—There seems to be no rational basis for determining the sizes of the steam dome. The diameters vary from 24″ for a 40 H. P. boiler to 32″ for a 100 H. P. boiler. The height of the dome is usually made equal to the diameter. Note the style of construction in Hutton, Figs. 362, 365 and 369. The latter is recommended by the H. S. B. & I. Co. When a dome is used it need not be stayed if the top is curved to a radius equal to or less than the diameter of the dome. A flat topped dome should be stayed.

Domes are used on locomotive and portable boilers, but are not especially to be recommended on stationary boilers. The dry-pipe is preferred by many to the dome because it weakens the boiler less and permits more compactness in design. A typical dry-pipe is shown in Fig. 25. The top of each leg L, L' is drilled with ¼″ holes such that the combined area of the holes is equal to or greater than the area of the delivery pipe. One row of larger holes (⅝″ to ½″) spaced 5 or 6 inch centers is drilled through the under side to drain the interior of the dry-pipe. The ends of the pipe are capped. The nozzle N may be of cast iron, cast steel or pressed steel. Cast iron and cast steel nozzles should be packed with a copper gasket when riveting to place, this copper is then swaged. The pressed steel nozzle B needs no copper gasket.

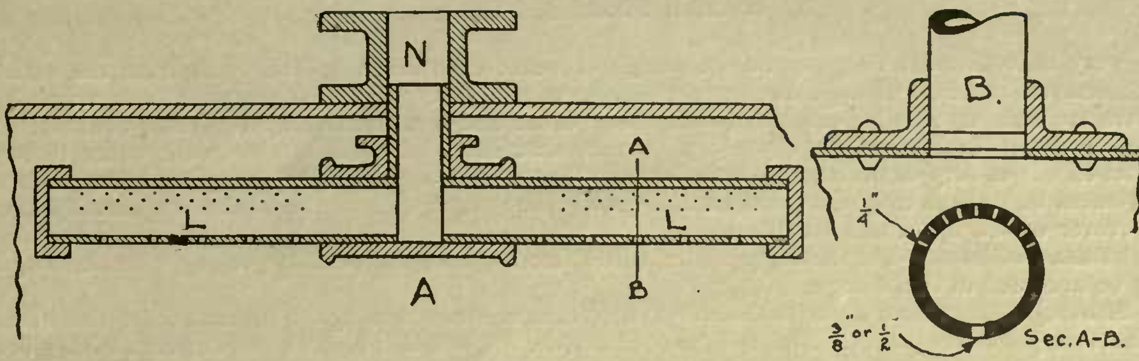


Fig. 25.

Steam Nozzle:—The formula for calculating the diameter of the steam nozzle is

$$A = \frac{V \times 144}{C}$$

where A = area of opening in square inches.

V = Vol. of steam in cu. ft. per minute passing through.

C = velocity of flow in F. P. M.

Ordinarily the velocity of steam is taken at 6000 F. P. M. This figure might be approached in short pipes or even in cylinder ports, but it is too large to use on a long steam pipe. The value of C should be between 2000 and 3000 for the steam nozzle. The volume of steam in cubic feet per second is

$$V = \frac{\text{H. P.} \times 30 \times S}{60}$$

where S = specific volume of steam at boiler pressure.

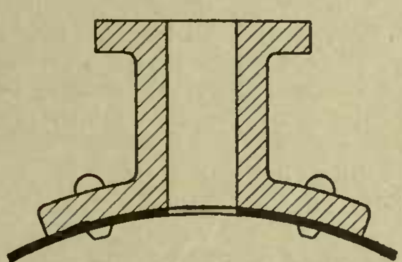


Fig. 26.

The size of the steam outlets should vary by half inches. A typical steam nozzle is shown in Fig. 26. When made of cast iron it should be designed very heavy. Cast steel is preferred.

Feed Pipe:—Figure the diameter d' from the same general formula as the steam outlet, using

$$S = \text{specific volume of water} = \frac{1}{62.4}$$

It is always well to have a large water inlet because of the tendency to lime up. After calculating the diameter by the formula, the next largest standard pipe should be taken. These figures are suggested for boilers not exceeding 150 H. P.

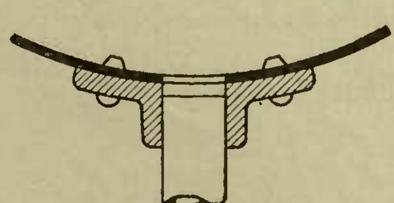


Fig. 27.

Enter the feed pipe into the boiler by flange fitting according to the H. S. B. & I. Co.

Blow-Off Pipe:—The following empirical formula is suggested for the blow-off.

$$d'' = \frac{\text{H. P.}}{60} + 1''$$

Attach the blow-off according to Fig. 27.

Very little data can be given on the design of water tube boilers. The types from the various manufacturers differ so widely that rules for design can be given only in the most general way. The boilers are built up in most cases with one lower drum and one or more upper drums connected by tubes as shown in Hutton Figs. 307-389. Water Circulates from one drum to the other through the tubes. It is very desirable therefore that the tubes have a good free passage and that their arrangement be such as to give a good natural draft. The water line of the boiler is generally about the center of the upper line of drums.

Steam Pressure:—The steam pressure varies from 150 to 200 pounds gauge. The latter figure may be accepted in this design.

Heating Surface and Grate Surface:—Water tube boilers are designed for 10 square feet of heating surface per horse power. This is measured on the fire side and refers to all surfaces having heated gases on one side and water on the other.

In the Stirling boiler this includes the tubes, all the lower drum and half the upper drum excepting the ends of each drum, while in the Wickes boiler it includes the upper part of the lower cylinder, all the tubes and about two-thirds of the surface in the upper cylinder.

After deciding upon the type of boiler to be designed the brick work must be planned around the steel work and the heating surface can then be proportioned to the various parts. The following ratios may be used:

Heating surface per H. P. .... 10. □′
Grate surface per H. P. .... .33□′
Ratio of H. S. to G. S. 30.3
Water evaporated per square foot of H. S. per hour from and at 212°F. .... 3.45 lbs.

Commercial Rating and Horse Power:—See same under Multitubular Boilers.

Materials:—See Multitubular Boilers.

Tubes:—Water tubes in a boiler are usually larger than the fire tubes in a multitubular boiler of the same capacity. The size most often used being 4 inches outside diameter. From 90 to 96 per cent. of all the heating surface is in the tubes, the latter figure applying to the Wickes boiler. Find the length of the tubes by locating the drums, and from this obtain the number of tubes.

Steam and Water Drum:—Find the thickness of the drum, as in figuring the multitubular shell, and plan for the least number of riveted joints.

In the horizontal type of boiler the drum ends are *dished* or *bumped* to a radius equal to the diameter of the drum, and do not need staying. In the vertical type the heads on both the upper and the lower drums should be stayed with through stays. The number of stays is estimated by allowing full pressure on all the head excepting the outer two inches which is supported by the shell.

Headers:—In the horizontal boiler the ends of the tubes are expanded into steel headers, which in turn make direct water connections with the drums. These headers are made of sheet steel and are thoroughly stayed. In the vertical boilers the tubes are expanded into the inner drum heads themselves.

Manholes and Handholes are placed in the ends of the drums in the horizontal type, or in the bottom of the lower drum in the vertical type. For sizes of plates, crabs, etc., see "Ryerson's Monthly Journal and Stock List."

Dry Pipe or Baffle Plate:—A dry pipe is well adapted to a water tube boiler, but a baffle plate is sometimes used instead. When a baffle plate is used it can best be located near the rear end of the upper drum so as to take steam from a point where it is free from currents and consequently contains less moisture.

The *steam nozzle* should be located near the center of the upper drum. For notes on nozzles and steam pipe see Multitubular Boilers.

Boiler Setting:—The headers in all water tube boilers either rest directly upon or are swung from I-beams which are built in the side brickwork. Where the header is above, a set of rollers is placed between the header and the I-beam to give easy expansion. Where it is below, the hanging is flexible enough to allow for expansion without interfering with the brickwork.

Baffle walls are inserted in the boiler setting to direct the gases along the tubes. Bolts, buck staves and fronts are used as in the average boiler setting.

Regulation in Boiler Design:—Boilers of the U. S. Navy are made according to government rules and regulations, known as "General Rules and Regulations Prescribed by the Board of Supervising Inspectors of Steam Vessels." These rules are printed in phamplet form and may be had for the asking. In Great Britain the same is under the control of the British Admiralty. The

German Lloyd line of steam vessels have their own rules. In stationary work, however, no such regulations exist and the proportioning of the boiler parts is left entirely with the designer. When the boiler is ready for use it is insured by some representative insurance company, and reinsured at stated intervals during use. The largest company in the United States is the Hartford Steam Boiler and Insurance Company, which publishes a paper called "Locomotive," containing a record of boiler explosions and other information valuable to the trade. Representative companies in England are the Vulcan Insurance Company and the Manchester Steam Insurance Association.

GAS ENGINE DESIGN.

(Arranged by C. S. JOHNSON.)

Problems:—Work up a set of problems on the same general lines as those under Steam Engine Design.

Indicated Horsepower:—The ratio of B. H. P. to I. H. P. in gas engines will be taken at .8 in the larger to .7 in the smaller engines.

Let P = M. E. P. in pounds per square inch for the working stroke.
L = length of stroke in feet.
a = effective area of piston in square inches.
N = R. P. M.
n = explosions or impulses per min.
S = piston speed in F. P. M. assuming velocity constant at all points of the stroke.

$$\text{Then I. H. P.} = \frac{P\,L\,a\,n}{33000} \tag{21}$$

$$n = \frac{N}{2} \text{ for four-cycle engines.}$$

n = N for two-cycle engines.

Equation (21) gives the I. H. P. for one cylinder (single acting). For other combination of cylinders and for double-acting engines proper factors must be introduced.

Piston Speed:—$S = 2\,L\,N$ or $L\,N = \frac{S}{2}$, Then (21) becomes $\text{I. H. P.} = \frac{P\,a\,S}{4 \times 33000}$ (22) for a single acting four-cycle engine.

and (21) becomes $\text{I. H. P.} = \frac{P\,a\,S}{2 \times 33000}$ for a single acting two-cycle engine. (23)

The following table gives values representing the limits of piston speeds used in practice and which vary slightly from steam-engine practice.

I. H. P.		$S = F.\,P.\,M.$			
1000 Stationary	700	to	1000	800	Average.
700 "	700	"	800	750	"
500 "	650	"	850	700	"
150 "	600	"	800	650	"
50 "	500	"	700	600	"
Small "	450	"	700	550	"
2-10 H. P. per cyl. (Automobile)	600	"	1000	750	"

Compression Pressure and M. E. P.:—High compression permits of easy ignition of weak gas which would not burn in the open air and this makes it possible to use a very weak fuel, hence giving more work from a given amount of fuel, that is, better economy. High compression also gives a correspondingly increased M. E. P. so that from the point of economy and power, high compression is to be advised.

Compression will vary with the fuel, speed, manner of cooling, etc., and is difficult to predict with any degree of certainty, however, a few fuels and their compression pressures obtained in practice will be given.

Gasoline.—Compression varies from 45#□″ to 95#□″ gauge, with an average of 65#□″. In automobiles the compression pressure is sometimes kept low to permit easy starting. In slower speed engines with better cooling, compression will vary from 60#□″ to 85#□″, with an average of 70#□″.

Kerosene.—Small engines running from 250 to 500 R. P. M. compression will vary from 30#□″ to 75#□″ and with independent vaporizers compression will vary from 45#□″ to 85#□″, with an average of 65#□″.

City Gas.—In small engines, compression varies from 60#□″ to 100#□″, with an average of 80#□″.

Natural Gas.—For medium and large engines with good cooling, compression ranges from 75#□″ to 130#□″, average 115#□″.

Producer Gas.—Compression varies from 100#□″ to 160#□″.

Blast Gas.—Used in large engines and takes compression at from 120#□″ to 190#□″, average 155#□″.

M. E. P.:—As the M. E. P. depends upon compression, mixture, etc., its determination involves an expression whose use is laborious. Fig. 28, gives the relation between M. E. P. and compression pressure directly without calculation.

Theoretical Indicator Cards:—Having determined the principal dimensions of the proposed engine the first thing to be done is to construct the Theoretical Diagram Sheet. As an illustration of these a set of four-cycle engine diagrams will be constructed. Fig. 30.

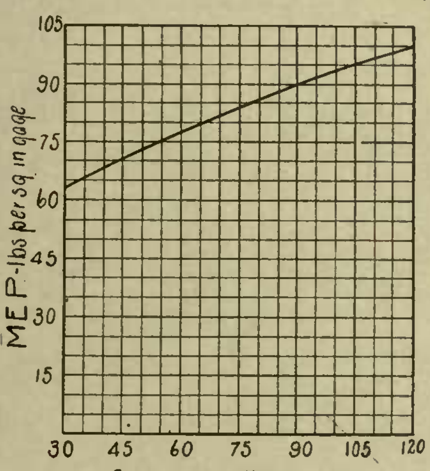


FIG. 28.

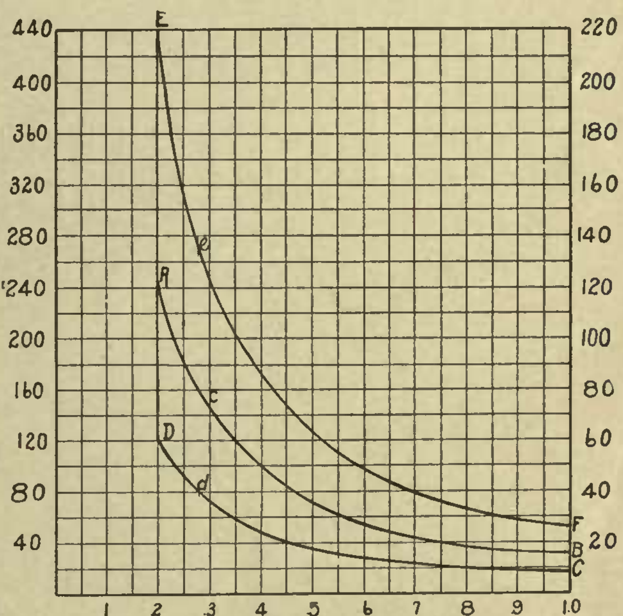
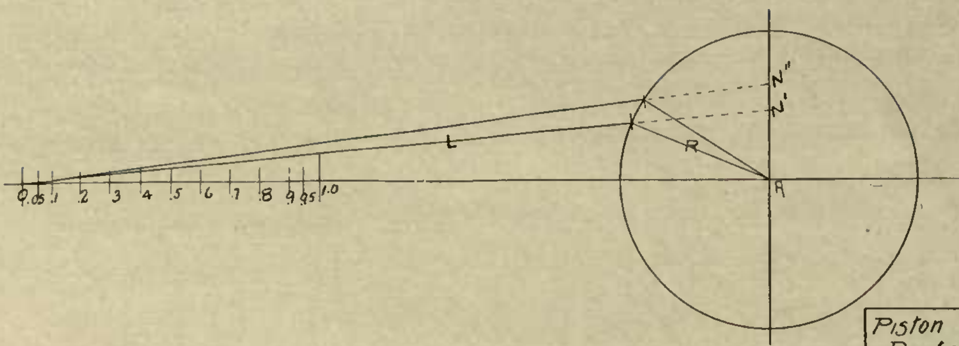


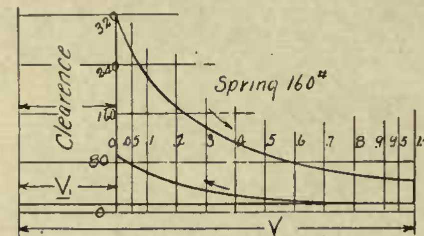
FIG. 29.

To do this one of the following factors must first be determined, namely, cylinder volume ratio or compression pressure; in order to make use of Fig. 29 in which the ordinates represent absolute pressures in pounds per square inch and the abscissae represent the total cylinder volumes.

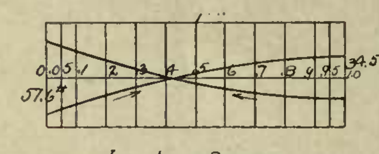
The curves *A B, C D* and *E F*, follow the law $P V^{\frac{4}{3}} = C$ which comes nearest the average practice.

The curve *A B* gives the relation between volume and pressure during compression and is read by the scale to the right of the figure. Curve *C D* is the same as *A B* but is plotted to the scale at the left. Curve *E F* gives the relation between volume and pressure after the charge has been ignited and is plotted to the scale to the left.



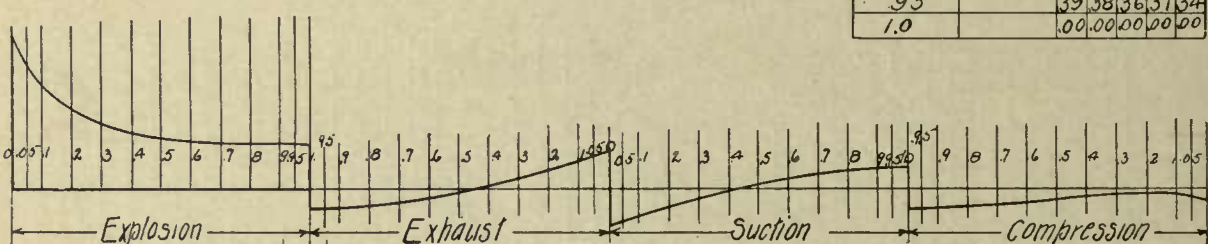


Theoretical Indicator Card.



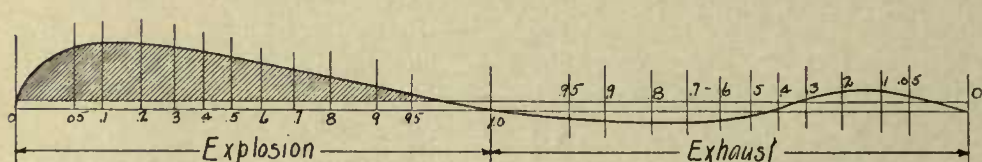
Inertia Diagram

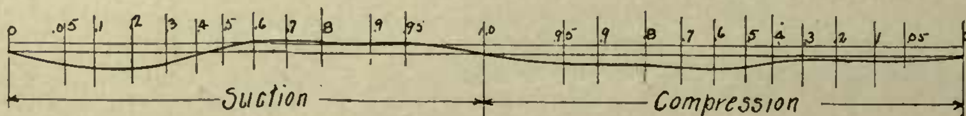
Piston Position	Values of $\frac{AN}{R}$					
	$\frac{R}{L}=$	$\frac{1}{6}$	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2.5}$
.00		.00	.00	.00	.00	.00
.05		.47	.48	.50	.50	.50
.10		.64	.65	.67	.68	.70
.20		.84	.86	.88	.90	.92
.30		.95	.96	.98	1.01	1.00
.40		1.01	1.02	1.03	1.06	1.08
.50		1.01	1.01	1.02	1.04	1.06
.60		.97	.97	.96	.98	1.00
.70		.88	.87	.85	.89	.89
.80		.76	.74	.72	.74	.74
.90		.54	.52	.50	.52	.50
.95		.39	.38	.36	.37	.34
1.0		.00	.00	.00	.00	.00



Wrist Pin Pressure Diagram

Tangential Pressure Diagrams





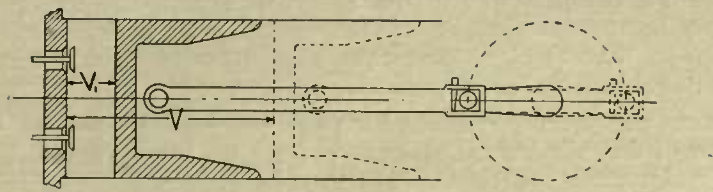


FIG. 31.

Volume Ratio is the relation of clearance volume to that of exhaust, that is, volume ratio $= \frac{V_1}{V}$ Fig. 31. As an example take $\frac{V_1}{V} = .28$ this gives a compression pressure of 80#□″ *abs* (Fig. 29 scale to right) and an explosion pressure of 270#□″ *abs* (scale to left). If the compression pressure was known and it was desired to find the volume ratio, the above process would be reversed.

The theoretical card would be formed then by the lines $C\ d$ (compression), $d\ e$ (explosion) $e\ F$ (expansion), and $F\ C$ (exhaust), all pressures being read by the lefthand scale. The area inclosed in this card represents the work done per working stroke.

Accelerating Force and Inertia Cards:—The accelerating force taken up and given out by the reciprocating parts each stroke, is given by the following formula for the development of which see Young's Steam Engine and Boiler Notes.

$$F = \frac{W\,V^2}{32.2\,R}\left(\cos\theta\ \frac{\sin^2\theta}{\sqrt{\frac{L^2}{R^2} - \sin^2\theta}} + \frac{L^2\cos^2\theta}{R^2\left(\frac{L^2}{R^2} - \sin^2\theta\right)^{\frac{3}{2}}}\right) \qquad (24)$$

where F = accelerating force.

θ = crank angle.

V = velocity of crank-pin in *F. P. S.*

W = weight of reciprocating parts in lbs.

R = length of crank in feet.

When $\theta = 0$ the crank is on head-end dead-center and equation 24 becomes

$$F = \frac{W\,V^2}{32.2\,R}\left(1 + \frac{R}{L}\right) \qquad (25)$$

and when $\theta = 180°$ the crank is on crank-end dead-center and equation 24 becomes

$$F = -\frac{W\,V^2}{32.2\,R}\left(1 - \frac{R}{L}\right) \qquad (26)$$

F will be equal to zero when the crank and connecting rod are at right angles, i. e., when the piston has attained its maximum speed. The distance the piston is from the head-end when at its maximum speed will be found as follows:

Let Q Fig. 32, be the piston position when R and L are at right angles and let

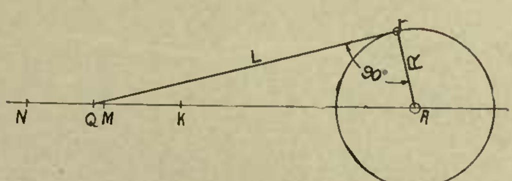


FIG. 32.

$$\frac{R}{L} = \tfrac{1}{4} \qquad R = \frac{NK}{2}$$

Then $N\,Q = A\,N - A\,Q$

$$= (L + R) - \sqrt{L^2 + R^2}$$

$$= 5\,R - \sqrt{17\,R^2}$$

$$= .44\,N\,K \text{ or } \frac{N\,Q}{N\,K} = .44$$

F in equation 24 is the total accelerating force in pounds necessary to accelerate the reciprocating parts but as the inertia curves are to be combined with the theoretical indicator cards whose ordinates are in pounds per square inch, the ordinates of the inertia curves should be in the same units.

Therefore if we divide both sides of equation 24 by a, the area of the piston, we get the ordinates of the inertia cards to the proper scale. To find the value of $F \div a$ in equations 25 and 26 involves a knowledge of $W \div a$, the weight of the reciprocating parts per square inch of piston area.

Fig. 33 gives the relation between the weight of the reciprocating parts per square inch of piston area and the diameter of the cylinder and by its use the weight of the reciprocating parts can be estimated if the cylinder diameter is known.

Fig. 33.

We are now able to find three points on the inertia diagram, viz. $F \div a$ for $\theta = 0$, $F \div a$ when $\theta = 180°$ and when $F \div a = 0$. These should be plotted as ordinates on a base line corresponding to the stroke, to same scale as theoretical indicator card. A circle struck through these three points will be a very close approximation to the true inertia curve.

In the Inertia Diagram, Fig. 30, $F \div a$ for $\theta = 0$ is $+ 57.6$ pounds and $F \div a$ for $\theta = 180°$ is $- 34.5$ pounds. $F \div a = 0$ at .44 stroke.

It will be noted that in this discussion the force that accelerates the reciprocating parts from 0 to their maximum i. e., when they are taking up energy, was called positive (+) while the force given out as the reciprocating parts are retarded was called negative (—). In order to accelerate the reciprocating parts this accelerating force must come from the gas pressure behind the piston and from the gas pressure standpoint must be viewed as a negative force and on the other hand when the reciprocating parts are being retarded they give out their stored up energy and exert a helpful influence and from the gas pressure stand point must be said to be positive. This will be made clear by a study of Figs. 34 and 35.

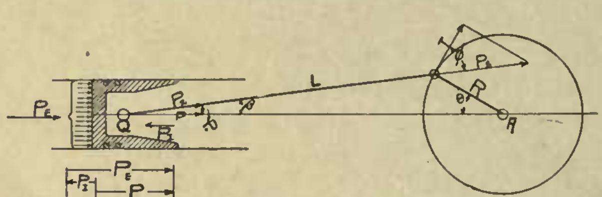


Fig. 34.

Let P_2 = Pressure along connecting rod. T = Tangential pressure producing rotation. $P_2 = \dfrac{P}{\cos\beta}$

$$T = P_2 \cos\Phi = P\frac{\cos\Phi}{\cos\beta} \qquad (27)$$

P_E = Explosion pressure.
P_I = Force necessary to accelerate reciprocating parts.
Then from Fig. 34, $P = P_E - P_I$ = Pressure on wrist-pin.
In Fig. 35, $P = P_E + P_I$

Wrist Pin Pressure Diagram:—The *theoretical indicator card,* Fig. 30, gives the gas pressure acting on the piston at any instant, and from the *inertia card* can be gotten the force of inertia acting at the same instant. Combining these we obtain the *wrist pin pressure diagram* giving the force at the wrist pin available for producing rotation of the crank. These are plotted to a four stroke base.

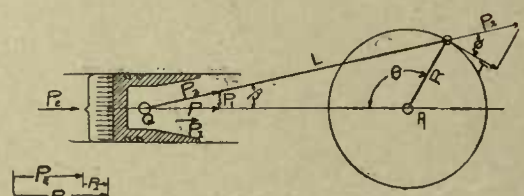


Fig. 35.

Tangential Pressure Diagram:—If we multiply the ordinates of the *wrist-pin pressure diagram* by the values of $A\ N \div R$ for the different crank positions corresponding to the proper piston positions and plot on a base circle whose diameter is equal to the stroke we obtain the *tangential pressure diagram.*

The values of $A\ N \div R$ for several values of $R \div L$ are given in the table, Fig. 30.

Rectified Tangential Pressure Diagram:—Changing from polar to rectilinear coordinates gives the *Rectified tangential pressure diagram.*

Mean Tangential Pressure:—Since the first or explosion stroke of the indicator diagram and the derived diagrams are all work diagrams their areas should be equal and proportional to the work done by this stroke.

Let M. E. P. = Mean Effective Pressure = P.
M. T. P. = Mean Tangential Pressure.
a = Area of piston in square inches.
L = Length of stroke in feet.

Since the M. E. P. for the theoretical indicator card is the M. E. P. for the working stroke only, the mean driving pressure for the four strokes or two revolutions is $P \div 4$. Then $P \div 4 \times 4 L$ is an expression for the work done in the cylinder and M. T. P. $\times\ 2 \pi L$ = work done at the crank-pin, but if we neglect friction, the work done in the cylinder is equal to the work done at the crank-pin.

$$\therefore P \div 4 \times 4 L = \text{M. T. P.} \times 2 \pi L \text{ or M. T. P.} = P \div 2 \pi. \qquad (28)$$

This M. T. P. is indicated on the *rectified tangential pressure diagram* by the ordinates of the long narrow unshaded portion. This diagram gives at ~~once~~ the relation between maximum and mean turning force, in this case *tangential pressure* (*max.*) $\div$ *M. T. P.* = 8.9 which is available for crank-shaft design.

It also gives the relation of the maximum energy delivered ΔE to the mean energy E which is a measure of the weight of fly-wheel necessary to limit the angular velocity variation during each cycle. This is the ratio of the shaded portion to the long narrow unshaded portion and in this case is $\Delta E \div E = 1.61 \div 1.68 = .96$.

ENGINE DIMENSIONS.

Cylinder:—The cylinder when not too thick is subjected to simple tension due to internal pressure which may be taken at 450#□″ max.

Length:—The length of cylinder will depend upon piston length, stroke, clearance, etc. Pistons usually project beyond their cylinders when single-acting and where short frames are desired.

Water Jacket:—All parts of cylinder exposed to the hot gases should be water cooled except on very small engines. The water-jacket should extend beyond the piston travel a short distance.
The amount of water should be sufficient to carry off 50% of the heat developed. Another method is to let the jacket take care of *two times* as much heat as is converted into work. Thus for a 25 H. P. engine the jacket should care for $\dfrac{25 \times 2 \times 33000}{778}$ B. T. U.

Water Space:—The water jacket should be free from pockets and of such shape that free circulation will result. Let w_1 = width of water space then $w_1 = .06\ d$ for horizontal cylinders. For vertical cylinders w_1 is less than this amount. For a thin swift stream w_1 will be made less, say $w_1 = .045\ d$ and for large cylinders $w_1 = .15\ d$.

Jacket Walls:—The jacket walls should be designed to carry an internal water pressure of 50#□″ also to support the parts of the mechanism and to take the weight of the cylinder when horizontal.
For small automobile engines, thickness of walls $t_1 = .06\ d$. For slow-speed stationary engines $t_1 = .10\ d$ and where reinforced $t_1 = .05\ d$.

Valves:—Valves are of two general types, flat and conical. Gases should enter and leave the cylinder with as little expenditure of energy as possible. To obtain the best results suction should take place as little below atmosphere as possible and exhaust should take place as little above atmosphere as possible. These pressures will depend upon the piston speed and valve opening. The valve lift should bear a constant relation to the piston speed in order that velocity of the gas through the valve be uniform.

Size:—The size of valve depends upon speed of the piston. The inlet-valve should be of such size that the mixture at atmospheric pressure should have a velocity of about 100 F. P. S. and exhaust-valves such as to give a velocity of about 85 F. P. S. For small high-speed engines these velocities will be somewhat greater.

Both inlet and exhaust valves are often made the same size, however, and vary in diameter from $\Delta = .3\ d$ to $.45\ d$.

Valve Lifts:—Valve lifts will vary from $h_1 = .05\ \Delta$ to $.1\ \Delta$ for flat valves and $h_1 = .07\ \Delta$ to $.16\ \Delta$ for conical valves, where Δ is the diameter of the valve opening in the seat, called valve diameter.

Thickness:—Valves when small are made of flat discs, when large are arched and are water-cooled. Flat valves are treated as plates supported at the edges and uniformally loaded for which Grashof gives

$$T = \sqrt{\tfrac{5}{6}\left(\frac{\Delta}{2}\right)^2 \frac{P_m}{f}} \qquad (29)$$

where T = valve thickness.

Δ = diameter of valve under unbalanced pressure.

f = fibre stress of metal.

P_m = maximum pressure in #□″.

Small engines have valve thickness of from $T = .09\ \Delta$ to $.23\ \Delta$ which corresponds to a fibre stress of from $f = 25\ P_m$ to $f = 4\ P_m$. The latter value gives a very stiff valve, one that will not spring or leak.

Flat valves have faces varying from $f' = .04\ \Delta$ to $.09\ \Delta$. Conical valves have faces of from 1.1 to 1.5 times width of seat. The normal pressure on cone-seated valves can be taken at $P' = \sqrt{2}\ P_v = 1.41\ P_v$ (for 45° valve) where P' = normal pressure and P_v = pressure on valve.

For small engines width of cone-seats vary from $.09\ d$ to $.15\ d$ while for larger engines the seats vary in width from $.06\ d$ to $.1\ d$. Outside diameter of valve should be large enough to permit the face to overlap the seat.

Valve Stems:—Exhaust valve stems must be strong enough to lift the valve against a terminal pressure of from 2.2 to 6.5 atmospheres. This latter being found in automobile engines of spark control type. For stationary work a value of 50#□″ can be used. Stems of a free length exceeding 15 diameters are treated as columns under flexure. The length of the stem bearing will vary from $1.4\ \Delta$ to $3.5\ \Delta$ Diameter of stems vary from $.22\ \Delta$ to $.28\ \Delta$

Valve-Closure Springs:—Helical coil springs are used almost universally. From 8 to 15 coils or turns are used on cam operated valves and from 5 to 6 on automatic valves.

Let W_s = force (assumed constant) necessary to close valve in a given time (t).

w = weight of valve.

h = valve lift in inches.

a = acceleration.

V = velocity in F. P. S.

S = space passed over.

The weight w must move through $h \div 12$, feet in t seconds.

$$V = \frac{S}{t} \qquad (30)$$

and

$$S = \tfrac{1}{2}\, a\, t^2 \qquad (31)$$

From 30 and 31 we get

$$V = \frac{h}{12\,t} = \frac{a\,t^2}{2\,t}$$

$$\therefore\ a = \frac{h}{6\,t^2} \text{ inches per sec}^2\text{. and } W_s = \frac{w\ h}{32.2 \times 6\,t^2} = \frac{w\ h}{193.2 \times t^2} \qquad (32)$$

or $t = .072\sqrt{h\,\dfrac{w}{W_s}}$ = time in seconds neglecting friction.

The spring should be capable of closing the valve in ¼ of a stroke or ⅛ revolution. This was found by Mr. H. L. Towne to be the maximum time required in a series of experiments with automatic valves on automobile engines.

Let N = R. P. M. then one revolution will take $60 \div N$ seconds of time and the time of closure of valve will be $t = 60 \div 8\,N$ or $t^2 = 56.25 \div N^2$ (33)
and from 32 and 33 we get for ¼ stroke closure $W_s = w\,h\,N^2 \div 10867.5$ (34)

For automatic or vacuum-opened valves the spring tension will vary from 6 to 30 ozs. per sq. in. of valve area with average of 12 oz. or .75 pounds.

For mechanically operated valves W_s should always be figured by equation 34 and should give values that will be between $W_s = 5.5\,\pi\Delta.^2 \div 4$ to $9\,\pi\,\Delta^2 \div 4$.

In the selection of springs there is a wide range of dimensions the only limitations being (1) a certain minimum load, (2) no practical change in load for the required valve lift.

Cams:—The cams should be so designed that the lift at all times is proportional to the piston-speed with such exceptions as are pointed out in Lucke's Gas Engine Design, Cuts 102 to 110.

Timing:—*Inlet-Valves should not open* until exhaust valves are closed. If exhaust closes on center or a little later the inlet-valves should open still later.

Engine.	*Inlet opens.*	*Inlet closes.*
Large slow-speed	about 5° after center	about 10° after center.
Medium or slow-speed small	about 3° after center.	about 4° after center.
Small high-speed	about 6° after center.	about 15° after center.

Exhaust Valves should open early. The amount of advance depends upon speed, diameter of valves, etc.

Engine.	*Exhaust opens.*	*Exhaust closes.*
Slow speed	about 25° before center.	On center.
Small high-speed	about 35° before center	2° after center.

Piston, Piston Heads:—Piston-heads are treated as flat plates supported at the edges and uniformally loaded, for which Grashof gives

$$T = d\sqrt{\frac{P_m}{6f}} \text{ or } f = \frac{d^2 P_m}{6T^2} \qquad (35)$$

in which $T =$ thickness of piston head in inches.
$d =$ diameter of piston in inches.
$f =$ fibre stress.
$P_m =$ max. pressure in cylinder.

Engines of diameter under 6 inches usually have flat unstayed heads varying from $T = .04\ d$ to $.08\ d$ with an average of $T = .06\ d$. Where lightness is a factor (automobiles) piston-heads as thin as ⅛ inch are successfully used. Engines of diameter over 6 inches should be web-stayed for stiffness. Where arched-heads are used the thickness can be reduced to $.6\ T$. Webs are usually $.6\ T$.

Length—In single-acting engines the wrist-pin is carried in the piston, this produces a rubbing on the sides of the cylinder due to the side thrust of the connecting-rod. The piston length must be sufficient to reduce this to a minimum.

Max. pressure on head $= P_m\, \pi\, d^2 \div 4$.
Max. pressure on guide $= P_1\, \pi\, d^2 \div 4$.
But $P_1 = P_m \tan \beta$ (max.) Fig. 35.
where $P_1 =$ Pressure on guides or between piston and cylinder.
$P_m =$ Max. pressure on piston.
$\beta =$ Angle between connecting-rod and center-line of engine.

If $L =$ length of piston and $d =$ diameter, then the projected area $= L \times d$ to receive the maximum guide pressure P_1.

Then
$$P_1 = \frac{\pi d^2}{4} \times \frac{P_m \tan \beta \ (max.)}{L \times d} \qquad (36)$$

In practice P_1 will run from 4 to 22#□″ and the length will run from $L = d$ to $L = 1.6\ d$ for high-speed engines. For small stationary engines $L = 1.4\ d$ to $2.3\ d$ and for large stationary engines $L = 1.2\ d$ to $1.7\ d$.

Piston Rings:—The number of rings vary 3 to 10. A clearance of about .001 inch is allowed at the sides to prevent sticking. They are usually made of C. I. being thin at the ends and thicker in the middle. Original diameter of rings before cutting is $= d + .175''$ for small rings, $d + .25''$ for rings for 8″ to 18″ engines and for very large rings an increase of $.01\ d$ is allowed. The width will vary from ¼″ to ¾″. The bridge between rings varies from ½ to 1½ times the width of ring.

Wrist or Piston Pin:—A pressure of from 750#□″ to 1200#□″ projected area should be used. Piston pins are figured in the same manner as wrist-pins in steam engines. (See Steam Engine Notes.)

Connecting Rod:—(See Steam Engine Notes). Automobile engines usually are circular and mid-section of, $d = .011\ D\ \sqrt{P_m}$ to $d = .014\ D\ \sqrt{P_m}$. When rectangular and of a thickness t and depth d. $d = 2.25\ t$ at crank and $d = 1.5\ t$ at wrist pin.

Cranks, Shafts, Pins and Cranks:—These will follow the design for similar parts in the steam-engine (see Steam Engine Notes). A pressure of 400 to 500#□″ can be allowed on crank pins and from 200 to 250#□″ on crank shaft bearings.

Fly-Wheels: *Steadiness of Speeds*—The most important function of the fly-wheel is to maintain an angular velocity within certain prescribed limits of variation throughout the *cycle*. This we will call the *steadiness of speed*. Steadiness of speed therefore is the variation of speed between one impulse and the next and is dependent only on the energy storing power of the fly-wheel. Steadiness of speed is *not* the per cent of difference of speed between no load and full load, a mistake often made; for if the no load speed was 220 R. P. M. and the full load speed was 212 R. P. M. Then 220 — 212 = 8 R. P. M. and 8 ÷ 216 = 3.7% which is *not* the coefficient of steadiness of speed. The revolutions per mniute are controlled entirely by the governor.

The rational method of calculating the stored-up energy of the fly-wheel is to make a graphical diagram of the changes that take place at the crank-pin due to the combination of gas and inertia forces and calculate the weight of fly-wheel from this consideration. Such a graphical diagram is the *Rectified Tangential Pressure Diagram*. Fig. 30 and the *coefficient of fluctuation of energy* is

$$K = \frac{\Delta E}{E} = \frac{\textit{area of shaded portion}}{\textit{area of long narrow unshaded portion}}$$

From Steam Engine notes we have for the weight of the fly-wheel rim $W = K\,E\,g \div n\,V^2$.
The values of n (*coefficient of unsteadiness*) for the different classes of work are given below.

For Pumping and ordinary duties	$n = .05$
For Driving machine tools	$n = .03$
For Driving textile machinery	$n = .025$
For Driving dynamos	$n = .02$
For Spinning machinery	$n = .01$
Marine Engines	$n = .15$
Automobiles	$n = .335$